USE OF INDICATOR DIAGRAMS IN STUDYING COMBUSTION IN A DIESEL ENGINE

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__Object__. The object of this investigation was to study the effects of various sizes of orifices in the fuel injection nozzles and pre-combustion chambers upon the combustion in a multicylinder compression ignition engine.

For purposes of comparison a series of economy runs were made and indicator cards and offset diagrams were taken under each condition. The effects of these changes were analyzed to show their influence upon the resulting combustion in the engine cylinder.

__Apparatus and Methods__. A marine type, vertical, four cylinder, four stroke cycle Hill Diesel engine, having a 5 inch bore and a 7 inch stroke was used during the investigation. This engine is of the compression ignition solid injection, antechamber type, having a compression ratio of 17.6.

The power was absorbed by a water-cooled prony brake having a beam length of 5.25 feet and a tare weight of 8 pounds. The beam load was measured on a Toledo springless scale. The power absorbing capacity of this brake limited the investigators to power loads below the rated power of the engine. Operation at high loads was so erratic that performance runs were limited to about 35 BHP.

Circulating water was furnished by the college mains leading into the circulating water pump attached to the engine, but dependence was not placed on the pump for pressure. The rate of flow of the circulating water was

controlled by a throttle valve ahead of the pump so as to maintain a circulating water temperature of 120 degrees F. at outlet.

Iron-constantan thermo-couples, manufactured by the Brown Instrument Company were installed in the exhaust passages between the exhaust valves and the exhaust header. The exhaust temperatures were balanced to within 20 degrees F. of each other and in addition the load balance between the cylinders was checked by removing the exhaust inspection ports and observing the relative intensity of sound of each exhaust.

In conducting this investigation it was decided to run the engine at 800 R.P.M., the rated full load speed. At this speed the engine operated more smoothly than at lower speeds. Prior to taking any data the engine was run for half an hour to allow thermal equilibrium to obtain. Tests were made starting with the lower loads, between runs time was allowed for the engine to warm up under the next load condition before the following run was made. Outlet circulating water temperature was maintained at 120 degrees F. throughout the investigation. Lubricating oil pressure was maintained at 75 pounds.

The injection advance device was not calibrated in such fashion as to permit advancing the injection by any predetermined number of degrees, but by means of a series

of holes drilled in a strip of iron which was then bolted to the engine housing it was possible to maintain any desired advance and return to the same setting at will.

Indicator cards were taken by means of the Bureau of Standards balanced diaphragm type of indicator.

This indicator (NACA #107 - Reference 1) consists essentially of the following elements: a water cooled pressure element, attached to and in communication with the engine cylinder space; a timer, connected on the end of the crankshaft and rotating at crankshaft speed; and the coordinating apparatus, consisting of source of pressure, gages for measuring pressure, and an electrical circuit for indicating balanced pressures.

The pressure element, Plate XXIII, was in this investigation installed in the relief valve casting, communicating directly with the combustion space. A series of small water-cooled holes conduct the cylinder gases to the under side of the steel diaphragm; the other side of the diaphragm is subjected to a pressure, by carbon dioxide, the amount of which is controlled and measured by the operator at the manifold. When the pressure on the cylinder side of the diaphragm exceeds that on the upper side, the diaphragm is forced against an insulated electrode in the top of the element, closing an electric circuit.

The timer, Plates XXIV and XXV, is in series with the electric circuit mentioned, and forms another break in it. A small copper insert in a bakelite disc makes contact with a stationary brush once each revolution for a brief interval. The arc of contact is one degree. The brush is adjustable throughout the entire circle, and its setting is read by means of a degree scale on the outer periphery.

The coordinating apparatus (7) consists of a manifold to which are connected three gages, one reading from 30" of mercury vacuum to 15 lbs. per sq. in. pressure, one, 0 to 100 lbs. per sq. in., and one, 0 to 1000 lbs. per sq. in. pressure, respectively. One end of this manifold is connected to a bottle containing liquid carbon dioxide, and the other to the pressure element. A vent and an aspirator connection complete the manifold. Adequate valves are installed to allow adjustment of any pressure in the manifold, and choice of gages suitable to the portion of the cycle being studied.

The electrical system is shown schematically in Plate XXVI. Two dry cells in series with a low resistance telephone receiver, the pressure element, and the timer form the essential parts of the circuit. When both the contact makers are closed, a click is heard in the telephone; another click is heard as the circuit is opened.

Refinements in the circuit consist of a variable condenser for varying the intensity of the click to suit the observer, and two switches by which either the timer or the pressure element can be short circuited to check the operation of the other.

The procedure was as follows:

1. With the engine running, and the indicator properly connected, the timer was set to any desired crankshaft angle.

2. The operator then listened on the phone. If a click of the same frequency as the speed of the shaft was heard, it indicated that the pressure in the manifold was less than the cylinder pressure for the particular point in the cycle for which the timer was set, and the pressure was increased in the manifold until the clicking ceased. The gage pressure at which the clicking ceased, i.e., when the diaphragm was just held away from the electrode was recorded as the cylinder pressure for that point.

3. The timer periphery is graduated in two degree increments, and points were selected around the cycle to give the desired spacing when plotted on pressure-volume coordinates. For the range from twenty degrees before top dead center to twenty degrees after top dead center, points were taken every two degrees for plotting of offset

cards of ignition and combustion.

4. For getting points on the suction and exhaust strokes, the pressure at which the double-click faded into the single click was recorded. Since the timer travels through 360 degrees, and the cycle lasts 720 crank degrees, points on the exhaust and suction strokes were on the same timer settings as homologous points on the compression and expansion strokes, respectively. When the manifold pressure was below that of the suction or exhaust line, as the case might be, the pressure element circuit was closed by the diaphragm once for each revolution of the timer, giving a pair of clicks for each revolution of the shaft, or a sound of twice the frequency of crankshaft speed (in the first case contact was made on alternate revolutions). As the observer becomes accustomed to the frequency of the normal clicks, he will notice promptly the doubling as occurring when passing through the pressure corresponding to the suction or exhaust point.

As noted in Reference 2, the points on the compression line have a very clear definition between clicking and not clicking, while on those points occurring during combustion and expansion, there is a range of pressures which will give a varying frequency of clicks, ranging from full frequency down to no clicks. This was due to uneven firing, uneven rates of burning, or, as suggested in

Reference 3 (NACA 396), due to small irregularities in injection advance angle from cycle to cycle. In this reference, the authors stated that oscilloscope readings showed a variation of ±2 degrees in this angle.

It was regarded as most accurate for a mean card, such as was being taken, to record the pressure at that point at which the clicks had been reduced to approximately one half their original frequency. This is the method used by the N.A.C.A. Laboratory in using balanced diaphragm types of indicators, (Reference 4).

The errors assumed as maximum were the result of opinions gained through several months use of the indicator. They are: timing,-plus or minus one degree; pressures,-on higher range plus or minus five lbs. per sq. in. The gages, of the Bourdon tube type, were carefully made test gages. A calibration of these gages showed no appreciable error in the two higher pressure gages, and an error within the limits of experimental and plotting errors in the low pressure gage. Since the comparison of the cards and offset diagrams was entirely on a qualitative basis, any calibration errors were of negligible effect so long as they were constant.

In arriving at the limits of accuracy in this investigation the authors have purposely chosen quite large tolerances for the variables under their control, preferring

to do this rather than to lead their readers to assume "point" accuracy and perhaps come to erroneous conclusions.

The speed of the engine was measured by a recently overhauled and calibrated tachometer, and checked at frequent intervals by a revolution counter. Checks showed that the actual speed of the engine was being maintained within 10 R.P.M. of the nominal speed. An error of 10 revolutions in 800 gives a maximum speed error of 1.25%.

The brake load was maintained within 1/2 lb. at low loads (25 lbs.) and within 3/4 lb. at the higher loads of about 40 lbs. From this the brake tolerance was taken as $\frac{.5}{25}$, or 2%.

The amount of fuel used per run was 1000 cc. This was measured by means of a sight gage alongside a stand-pipe which had a length of 18 inches. The level of the fuel in the sight gage could be read to within 1/10 in. giving a maximum possible error of 1/10 x 1/18 x 1000cc or 5.55cc. The percentage error would be .55%.

The time error should be negligible.

From these individual errors the following limits of accuracy were imposed upon the brake-horsepower and the fuel rates.

$$B.\ H.\ P. = \frac{R.P.M.\ x\ brake\ load}{x\ constant}$$

$$Brake\ H.P.\ error = \frac{1.0103\ x\ 1.02}{x\ constant} = \pm\ 3\%$$

$$Fuel\ Rate = \frac{lbs.\ fuel}{B.H.P.\ hrs.}$$

$$Fuel\ Rate\ error = \frac{1.0056}{1 - .03} = 1.037\ or\ \pm\ 3.7\%\ max.$$

By using the tolerance found above the authors felt that all their data would fall within these limits.

TEST RESULTS AND DISCUSSION

Effect of Varying Spray Orifice Diameter. In the following series of runs, all conditions were maintained constant except for the injection nozzles, and, in the runs indicated, the timing. In all runs, the #1, or original precombustion chamber was used, and for purposes of comparison of pressure-time cards, a uniform brake load of 31.6 B.H.P. was maintained. The original nozzle, having a .020" orifice was taken as standard. Its offset card is shown on Plate IX, and its brake economy is shown as a full line in all economy curves.

The first nozzle tested had a .010" diameter orifice.

The effect of this nozzle on brake economy is shown by curve 1, Plate VI. The effect on combustion is shown in offset card #7-A, Plate X.

From a study of the latter, in comparison with the standard card, the following was noted: (a) the beginning of the slow pressure rise, or ignition period, was advanced slightly, and the rate of pressure rise during the ignition period was increased; (b) the beginning of the period of rapid pressure rise, or inflammation period, was advanced by a crank angle of two degrees (.000417 sec.) and the rate of pressure rise during inflammation was reduced; (c) the full indicator card of this run, Plate V,

shows that burning extends well through the expansion stroke, giving a very heavy pressure and high temperature at exhaust valve opening. This was also observed in the unusually high exhaust temperature as measured by pyrometer, and by visual observation of flame passing through the exhaust valve.

In the authors' opinion, the advance of the injection and the increase in the rate of burning was due to two causes. The principal cause being a purely mechanical one, brought about by the higher pressure in the injection system at closure of injection valve. This furnished a higher initial pressure in the system before the start of the next stroke, and so reduced the injection lag by reducing the amount of compression of oil, and expansion of pipe lines before the injection valve was lifted. The secondary cause was the more rapid heating of the fuel particles as they entered the precombustion chamber due to the finer subdivision. This finer subdivision, which was produced by the larger pressure built up in the injection system, offered a much larger total surface to the vaporization and subsequent ignition.

In the inflammation period, the advance in timing noted was due probably to the mechanical advance mentioned above. The decreased slope of the pressure line was due

however, to (a) a slower burning because of uneven mixture, and (b) the slower rate of admission of the fuel. In this nozzle, the diameter of the solid jet was insufficient to provide penetration, and the type of the precombustion chamber was not conducive to turbulence. The result is that there was a very rich, slow burning mixture in the upper half of the chamber, surrounding the nozzle, and a very lean mixture in the lower end surrounding the passage to the cylinder. The fire, therefore, resolves itself into a very thin leg, and gradually increasing thickness as it reaches the toe. As has been stated, there was evidence that burning continued after exhaust. (Plates XXI and XXII also substantiate this.) Mr. W. F. Joachim suggested that this was due to the distribution of fuel in the precombustion chamber; conditions were such, with the over-rich mixture at the top of the chamber, and the necessary combining air outside the chamber, that a considerable portion of the fuel would never come into contact with sufficient air to form a combustible mixture.

In the light of these results, it was to be expected that the thermal efficiency would suffer, as the combustion was taking place through an ineffective portion of the cycle, (Reference 5). That such was the case was

shown by the comparison of the brake economy curve with that of the standard conditions.

In Plate XIV is shown a run on the same nozzle with full injection advance. An attempt was made to advance the injection until no flame appeared in the exhaust. This could not be done even with full advance. The card shows that combustion began about ten degrees before top dead center, but the exhaust pressure and temperature did not differ appreciably from that obtained with normal injection advance. This indicates that combustion lasted more than 150 crank degrees, (and hence injection possibly lasted that long). The economy was very poor, being only slightly better than that obtained from the same nozzle with normal advance.

The next nozzle tried was one having an orifice diameter of .030". All other conditions were maintained the same. The effect of this nozzle on combustion is shown in Plate XI.

A comparison of this offset card with the standard shows that the timing and rate of pressure rise during the ignition period were the same as that of the standard. However, the commencement of the inflammation period was delayed considerably, and the rate of pressure rise during the inflammation period was increased. The amount of the delay was 11.2 crank degrees, or .00275 seconds.

This delay may be almost entirely charged up to increased injection lag due to the large area of the orifice bleeding down the injection passages before the end of the discharge stroke. This is shown in Reference 6. In this investigation, conducted at Langley Field, the authors show pressures in the injection system during the pump discharge for nozzles of .020, and .030", respectively, for the same pump speeds. In general, the curves show the pressures with the .030" nozzle to be less than half those with the .020" nozzle, and the final pressure in their particular case to be about 3000 lbs. per sq. in. for the latter case, as against 700 lbs. per sq. in. for the .030" nozzle. No means were available of measuring the fuel pressures built up in the nozzles under investigation, but it may be assumed for comparison, that the pressures in the two cases were in approximately the same proportions.

The more rapid rate of pressure rise during inflammation was explained by the more rapid rate of injection, the fact that there was more fuel in the chamber before inflammation took place, due to the larger particles, and the more favorable distribution of the spray. In this case, the heavy, solid spray is sent down toward the cylinder passage in such manner that the ignition of the finely divided "border" spray near the nozzle will blow

the overrich mixture out into the cylinder to unite with the air there to form a combustible mixture. The economy obtained from this nozzle, while much better than that obtained from the .010" nozzle, was not as good as that obtained from the standard nozzle.

The same nozzle was then tested with the injection more advanced. The amount of advance was not definitely known, but by an arrangement of pin and holes, the position was fixed, and was repeated for all other runs of "advanced injection".

This advance (Plate VII) brought the beginning of the inflammation period up to three degrees before top dead center,--an advance of about 21 degrees (.[illegible] sec.) over the same nozzle with normal advance, or about 9 degrees (.[illegible] sec.) advance over the standard conditions. Rate of pressure rise during inflammation was judged to be about the same, considering its rise above the non-firing compression-expansion curve. The maximum pressure in this case was increased to 753 pounds per square inch as compared with 640 for the normal advance. As would be expected from the offset card, the brake economy was improved (Curve 3, Plate VI) over that obtained with the normal injection, but was slightly poorer than the original economy. Exhaust temperatures were not sensibly different from the same nozzle at normal advance.

The sound of the engine was noticeably louder and rougher than for any other conditions tested. A maximum brake horsepower of 63.2 (BMEP 77.6; IMEP 111.6) was reached, but at this load there was considerable knocking. This was partially due to the advance of timing with increased load, an inherent mechanical feature of the fuel pump, which will be discussed later.

Acting on the theory, derived from the first two trials, that the fine atomization of the .013" nozzle, to give rapidity of ignition, combined with an increase of area to give more speedy injection were desirable, the operators next installed a nozzle having five .013" diameter nozzles. This gave an increase of area of twenty-five percent over the original nozzles. The pressure-time card is shown on Plate XIII. Inasmuch as a few trial runs with normal injection showed late inflammation, and comparatively poor results, the recorded runs were made with advanced injection.

The increased lag in this case over the standard condition was charged, probably, to distribution lag. The penetration of the fine spray was estimated to be about one-half the length of the precombustion chamber. This would give a condition of approximately one-fourth of the total combustion volume being filled with an overrich, and therefore slow burning mixture. The region

of combustible mixture would be a surface on the cylinder side of the spray mass, and, therefore, instead of the combustion pressure tending to force the burning mass into the cylinder, it tended to force it back against the nozzle. The effect of this was regarded as important. Mr. R. D. Hill (Reference 11) states that investigation at the Hill Diesel Engine Company showed the pressure differential between the precombustion chamber and cylinder ranges from 100 to 150 pounds per square inch. This was found by taking simultaneous cards on the precombustion chamber and cylinder. This pressure, if generated behind the mass of the fuel has an excellent effect on distribution.

With advanced injection, economies slightly poorer than the standard conditions were obtained. The engine seemed to be constant and smooth in its operation. A maximum load of 46.2 brake horsepower (BMEP 85.9; IMEP 135.8) was obtained by dropping the injection advance back to the normal position. This retardation was necessary to counteract the inherent injection advance with increase of load, as mentioned above.

On all offset card and economy curve plates has been shown a full size sketch of orifice size and arrangement in the precombustion chamber and spray button used for that run.

EFFECT OF VARIOUS PRECOMBUSTION CHAMBERS ON COMBUSTION

The various precombustion chambers used in this investigation are shown in Plate XI. A brief description of them follows:

#1. This was the precombustion chamber previously installed on the engine, and was taken as representing the standard for comparison. The orifice between the antechamber and cylinder was 5/16" in diameter, and was sharp-edged.

#2. This antechamber had three orifices, arranged symmetrically about the center, as shown in Plate XVI. The diameter of each orifice was 1/4". Orifices were sharp-edged.

#3. This precombustion chamber had a single 9/16" diameter orifice, as shown in Plate XV. The orifice was sharp-edged.

#3A. This was the #3 chamber with the exception that the orifice had been flared out on both sides to form a rounded entrance orifice.

#4. This chamber had an orifice of 7/8" diameter. This practically eliminated the effect of the precombustion chamber.

The performance with #1 precombustion chamber has been discussed in the previous section as it was used in all the tests with varying spray nozzle diameter. In

the following investigation, the .030" diameter nozzle was used throughout, and the load and speed were held constant as in the previous section.

The pressure-time card for #2 precombustion chamber is shown in Plate XVI.

The effect on combustion of this precombustion chamber was to increase slightly the rate of pressure rise during the ignition period, and to delay the beginning of inflammation 1.6 degrees (.00036 sec.). The engine operation was very rough, and its load carrying capacity was very erratic. Great difficulty was experienced in keeping the load and speed constant. The only explanation suggested by the authors for this was that the arrangement of the antechamber orifices with metal in the center, breaks up the center of the oil spray, and disrupts the heavy oil jet, which for optimum conditions should carry into the cylinder. The economy varied slightly from that under the original conditions.

The next precombustion chamber tested (#3) had a single 9/16" orifice. The offset card of this chamber is shown in Plate XV. In this test, the beginning of inflammation was about the same time as in the original conditions, but the rate of pressure rise during inflammation was much more rapid. Operation of the engine seemed very smooth and regular. The economy showed an improvement

over any previous run, especially on lighter loads. The authors hesitate to advance any theory as to the performance of this chamber. It was felt that there must be an optimum size for this orifice, but the number of variables entering into this problem makes it difficult to determine. The size of this orifice is a function of: cylinder and antechamber dimensions; the relative volumes of each; the percentage of combustion occurring in the antechamber, and hence the volume of gases to be passed through it; the pressure differential desirable through the orifice; and many others. This particular size was chosen, because it was found at the N.A.C.A. Laboratories at Langley Field, (Reference 3) that for their test engine with identical cylinder dimensions, and same distribution of combustion space between cylinder and antechamber, that this was the best size. The shape of the precombustion chamber, however, was radically different.

In order to study the effect, if any, of the degradation of heat into kinetic energy due to high gas velocities through the antechamber orifice, an attempt was made to reduce this velocity by increasing the coefficient of discharge. This was accomplished by flaring out both sides of the #3 precombustion chamber (#3a) and converting it from a sharp-edged orifice to a convergent-divergent nozzle. Slight difference could be noted in

the pressure-time card (Plate XVII). The rate of pressure rise during combustion was reduced. The economy, however, on light loads was further improved. It is believed that there was an appreciable reduction in friction horsepower due to this change, because there was an improvement in economy which was more marked at the lower loads.

Precombustion chamber #4 caused a greatly increased ignition lag. For comparable results, data was recorded with the timing in the advanced position. The offset card is shown in Plate XVIII. The size of this orifice was such as to practically eliminate the precombustion feature of the engine, and to convert this space into cylinder volume. This would reduce the velocity of the jet of gas and fuel emerging from the antechamber, and, while the loss to kinetic energy would be less, the loss due to lower velocity and accompanying turbulence in distributing the charge through the cylinder combustion space was greater. That this was the case was indicated by the increase in lag (increase in distribution and spray penetration lag) and the poorer economy resulting.

Time was lacking to pursue the subject of precombustion chambers, further, but it was felt that the optimum size lies in the neighborhood of 9/16" diameter, rounded entrance orifice.

Some difference in performance may have been accounted for in the slight change in design of precombustion chambers Nos. 2, 3, and 4. These were furnished by the Hill Diesel Engine Company, and were made to a modified design. It had been thought that the original precombustion chambers on their engines of the type of #1, were operating at too cool a temperature due to the complete water jacketing surrounding them in the cylinder head. To increase this temperature, an attempt was made to reduce the heat dissipated to the cooling water by increasing the depth of the undercut on the side. This can be seen in the photograph (Plate II). Temper colors on the periphery of the later types of chambers indicated that their operating temperatures were higher than the original.

In general, it was found that economy was rather insensitive to wide changes in precombustion chamber design, although other operating characteristics, such as lag, noise, and evenness of operation were more sensitive.

<u>Effect of Load on Injection Advance Angle</u>. Plate III shows the marked effect of load on commencement of inflammation. In general, the lighter the load, the greater the delay in ignition. This was found to be due to the design of the fuel metering system. The method of metering is to vary the opening of a conical seated

needle valve on the suction side of the fuel pump. When the engine is throttled down, a greater quantity of oil is allowed to leak back before the commencement of delivery, and hence delivery occurs later in the pump stroke. Since the cam action is constant, this causes later injection in the cylinder. To offset this effect, all pressure-time cards recorded were taken at uniform loading.

Plates XX and XXI show the pressure-volume relationships plotted on logarithmic coordinates.

Plate XX is the diagram for the standard .020" spray nozzle. The slope of the compression line is about 1.34, while the slope of the expansion line varies from 1.03 in the early part of the expansion after rounding the top of the curve to 1.2 during the last half.

There is no fixed value in the opinion of the authors that may be assigned to the exponent of the expansion line that could be taken as typical of a good engine. Since the rate of injection and rate of burning vary, it is possible that a number of values may be acceptable. However, it is quite evident that in a Diesel cycle it is preferable that the exponent exceed unity, as a value of less than unity would indicate that the rate of generating the heat of combustion exceeds the net loss and the work done, thus causing the temperature to rise in the cylinder as it approaches the end of the stroke.

Plate XXI, the diagram for the .010" nozzle, shows this defect. In this case the slope of the expansion line is .9 and, as indicated in the temperature entropy diagram, (Plate XXII), the temperature is highest at about the point of exhaust.

Plate XXI shows a more inefficient cycle than would probably be obtained with an air engine having a cut-off near the end of the stroke. It has taken very little advantage of the expansion properties of the working fluid.

The authors, however, do not feel that too critical an analysis should be made of this type of experimental curve. The slight inaccuracies inevitable in taking indicator cards will give erratic and misleading information if this analysis is carried to extremes.

Plate XXII shows the temperature-entropy relationship for the cycle. The curve in black shows this relationship for the standard [illegible] in orifice while the curve in red is for the .010" orifice.

In constructing these curves, it was necessary to arrive at some figure for the mass of gases in the cylinder.

In arriving at the value of .00414 lbs. of fluid in the cylinder, the authors availed themselves of three methods for finding the mass, viz.:

1. $\text{Mass} = \dfrac{\text{Piston V} \times \text{air pressure} \times \text{Volumetric eff.}}{\text{abs. temperature} \times \text{gas constant}}$

2. $\text{Mass} = \text{Mass of fuel/cyc.} \times \text{air fuel ratio}$

3. $\text{Mass} = \dfrac{\text{Pressure at end of compression stroke} \times \text{Vol.}}{\text{gas const.} \times \text{temp. abs.}}$

By taking the average of the masses found by these three methods a value of [illegible] was obtained.

For purposes of illustration it is not important that the exact value of the mass within the cylinder be determined, since an approximate value will give the general shape and show the tendencies with equal clearness. The only difference would be one of position and it was believed that the value used gives this position with sufficient fidelity for this purpose. As an additional check, the value of the maximum temperature found by computation using the method illustrated by Professor H. A. Everett in his paper "The Prediction of Maximum Cylinder Temperatures Attained in Actual Internal Combustion Engines" checked quite closely with the values shown.

The temperature-entropy diagram sheets were constructed by Professor H. A. Everett of The Pennsylvania State College and are corrected for the effect of the variation of specific heat with temperature.

Plate XXII shows that in the early part of the compression stroke both curves approximate an adiabatic compression, and both show loss of entropy due to cool-

ing towards the end of the compression. From this point on, the curves are quite dissimilar. That of the .020 in. nozzle reaches its maximum temperature at much higher pressures than the .010 in. nozzle and at a point considerably earlier in the stroke. It is inherently of greater efficiency since it is making more use of the temperature rise.

The .010 in. nozzle, on the contrary, is doing its major work over a lower average temperature and then uneconomically discarding its heat at almost its peak temperature.

In the case of the .020" diameter nozzle, it is seen from data sheet # 5 that the maximum temperature was first reached at a point about 30°-40° past top dead center.

This card was also analyzed by the tangent method of Dr. P. H. Schweitzer (Reference 9). While the complete analysis of this card was not regarded as consistent enough to be recorded, it is believed significant that the point of maximum temperature as indicated by that method occurred at completion of 12% of stroke, or about 37° after top dead center. This checks the temperature-entropy diagram very closely.

Another interesting correlation lies in the fact that the slope of the log P - log V plot of this card

(Plate XX) at this point of maximum temperature is unity, which ensures the tangent analysis of the point $x = 1$.

CONCLUSIONS

Spray nozzle orifice size has an appreciable effect upon the combustion in a Diesel engine cylinder. The optimum size of injection nozzle orifice in the engine under consideration was .020 in. Smaller orifices decreased the rate of burning causing combustion to last up to the point of exhaust valve opening; larger orifices increased the injection lag and hence caused a delay in the start of combustion but a more rapid burning ensued.

Precombustion chamber orifice size has small effect upon the economy especially at brake loads near the rated horsepower. Several characteristics, such as quietness, smoothness of operation, regularity of firing, which could not be evaluated in laboratory tests, but which are important to the builder and operator were affected. In general, the best results were obtained with the 9/16" rounded entrance orifice antechamber.

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XIX Effect of load on lag.

XX Log P vs. Log V, Normal Card.

XXI Log P vs. Log V, .010" Nozzle.

XXII Temperature-Entropy Diagram.

XXIII Pressure Element, Balanced Diaphragm Indicator.

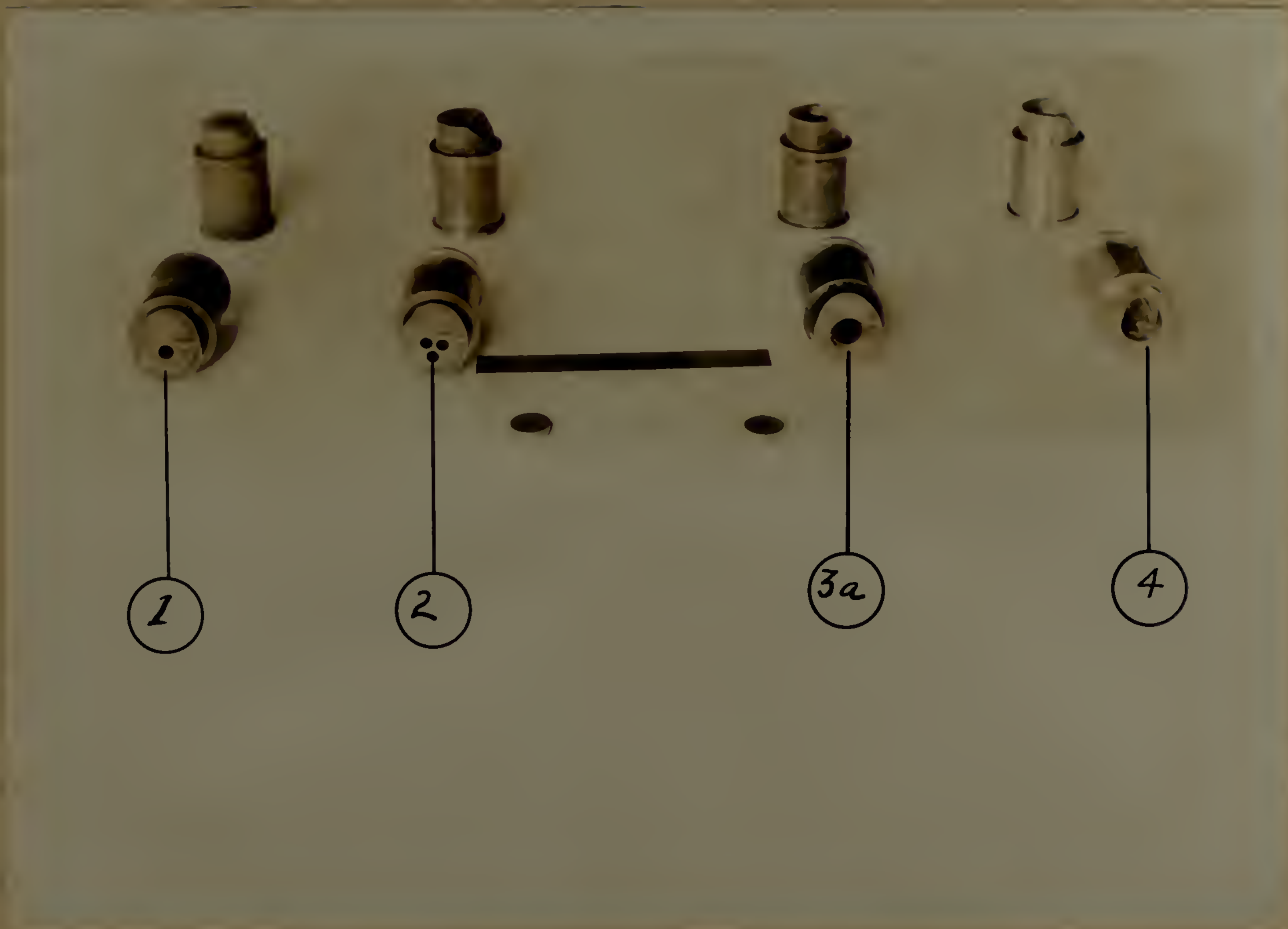
XXIV, XXV Timer, Balanced Diaphragm Indicator.

XXVI Schematic Diagram of Balanced Diaphragm Indicator Electric Circuit.

[illegible]

[illegible]

PLATE I



PRECOMBUSTION CHAMBERS AND SPRAY BUTTONS

PLATE II

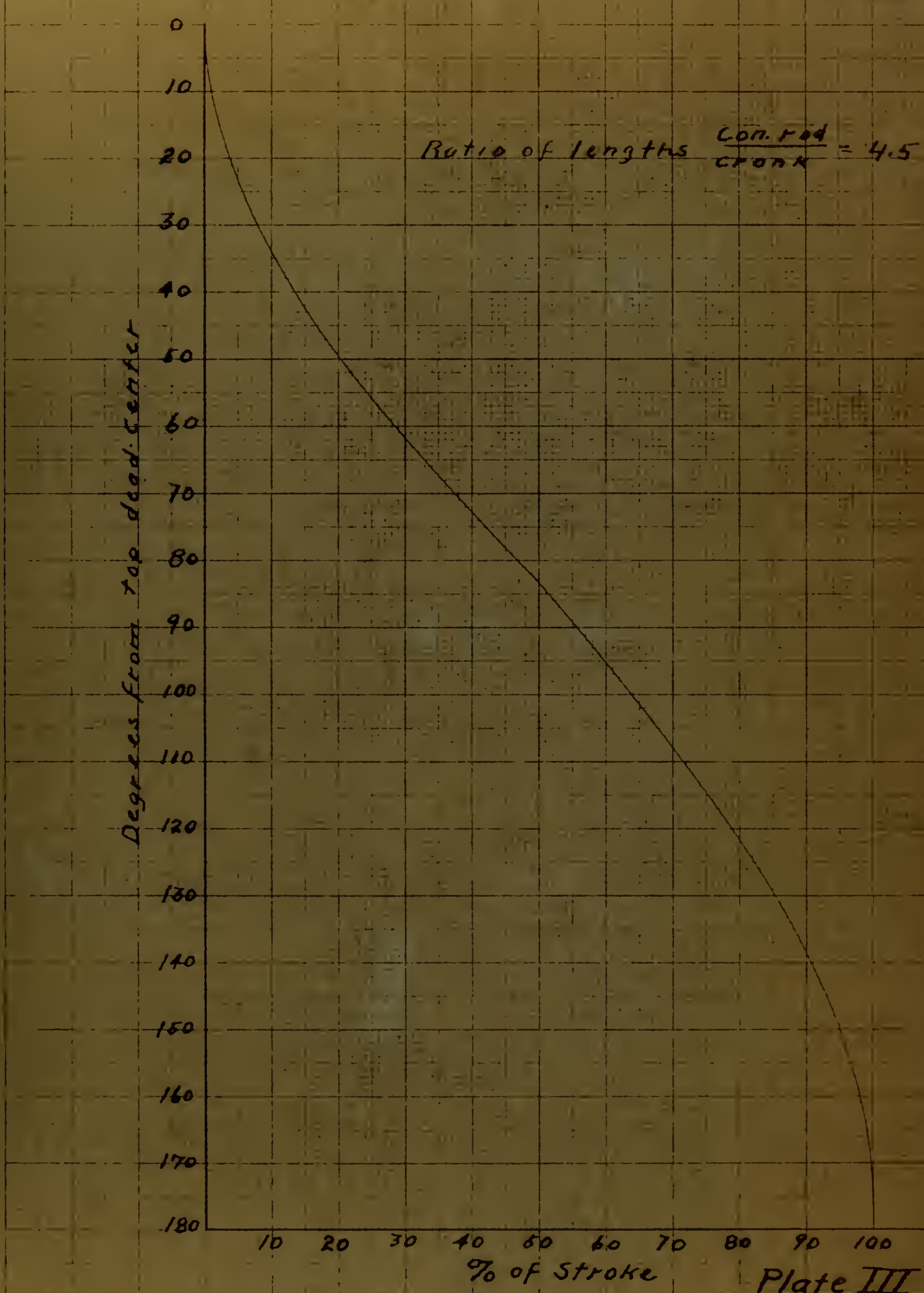

Ratio of lengths $\frac{\text{Con. rod}}{\text{crank}} = 4.5$
Degrees from top dead center
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
10
20
30
40
50
60
70
80
90
100
% of Stroke

Plate III

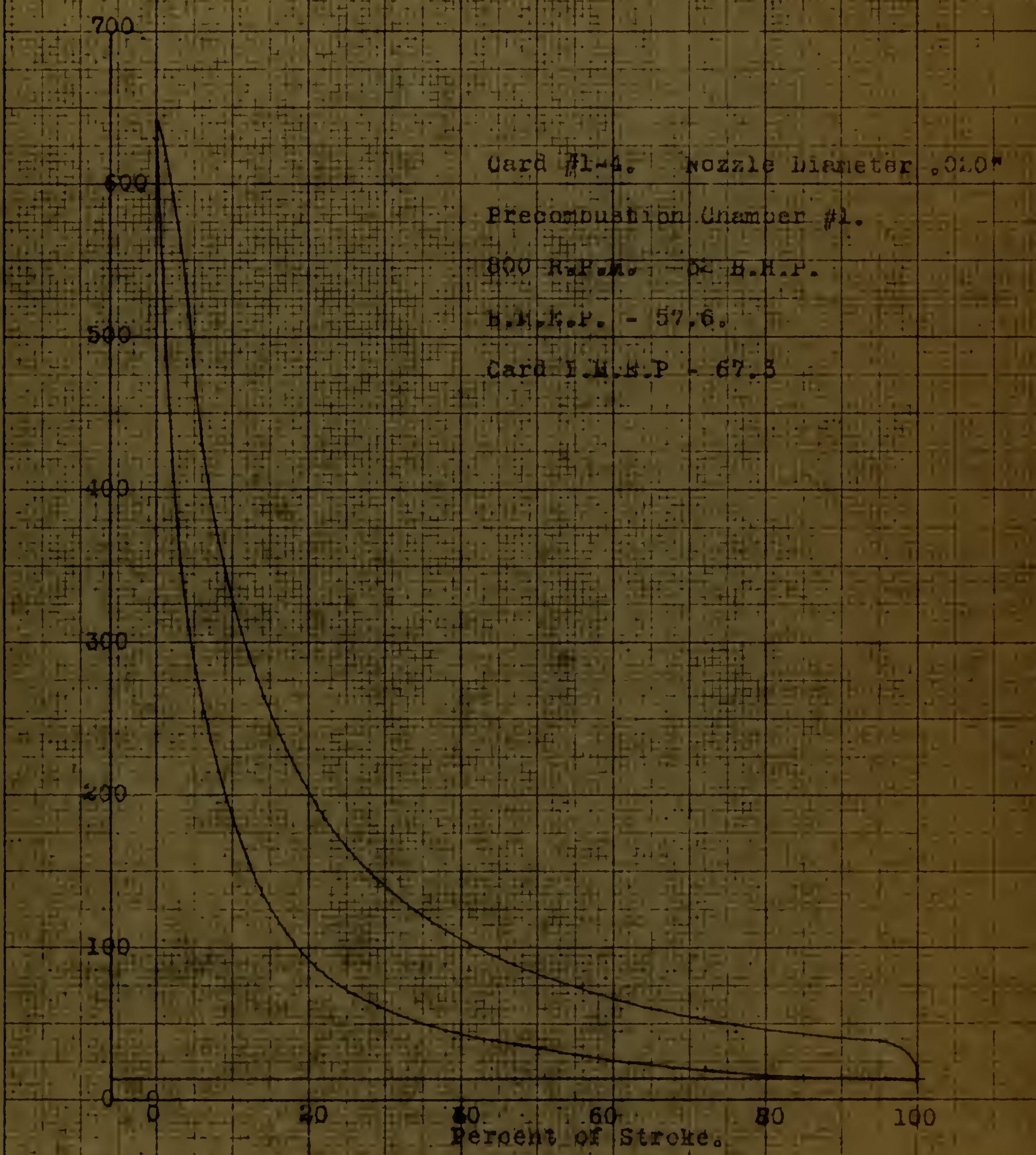
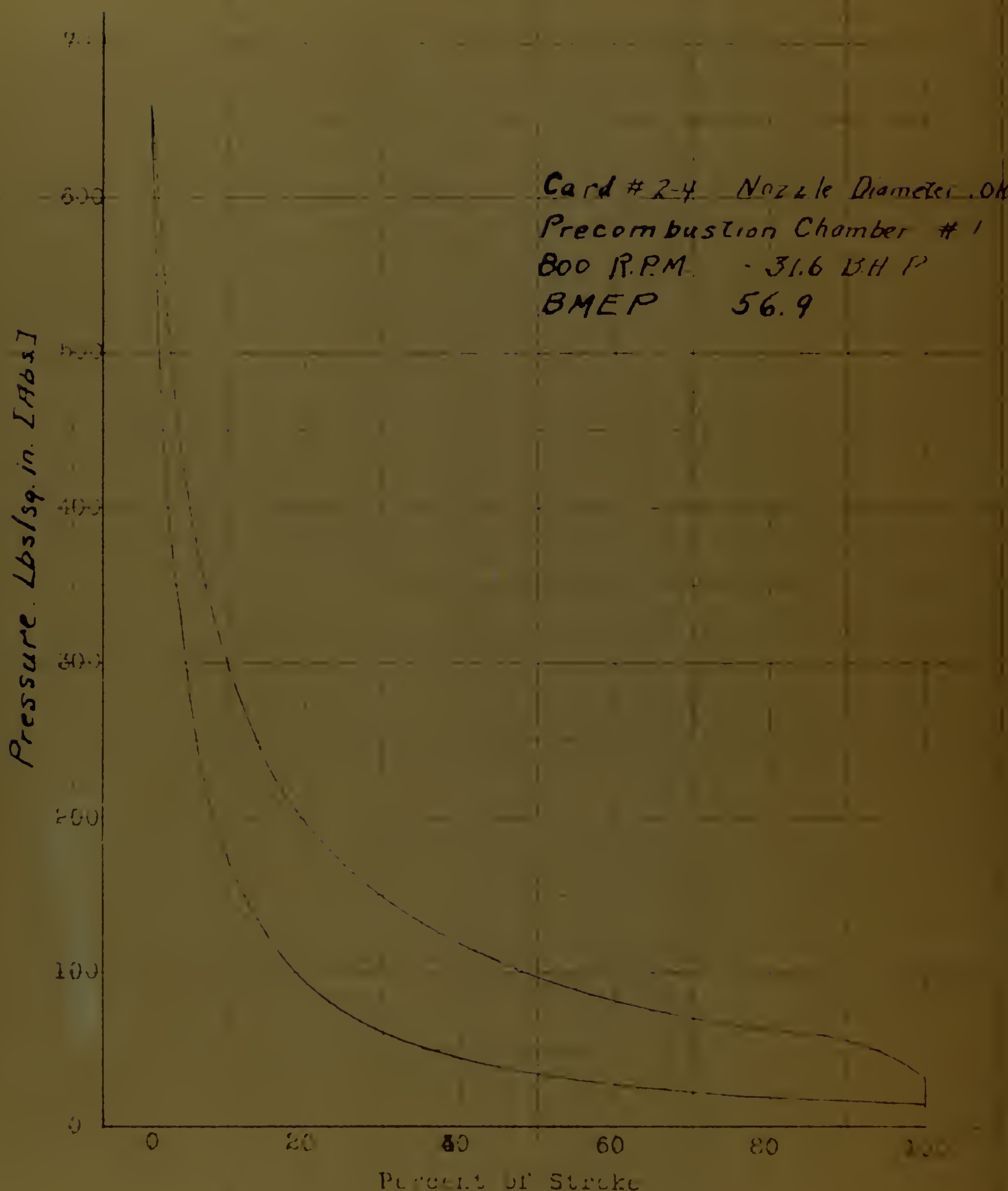

Card #1-4. Nozzle Diameter .020"
Precombustion Chamber #1.
800 R.P.M. - 52 B.H.P.
I.M.E.P. - 57.6.
Card I.M.E.P - 67.3
700
600
500
400
300
200
100
0
0
20
40
60
80
100
Percent of Stroke.

PLATE IV



Card # 2-4 Nozzle Diameter .010
Precombustion Chamber # 1
800 R.P.M. - 31.6 B.H.P.
BMEP 56.9
Pressure Lbs/sq. in. [Abs]
600
500
400
300
200
100
0
0
20
40
60
80
100
Percent of Stroke

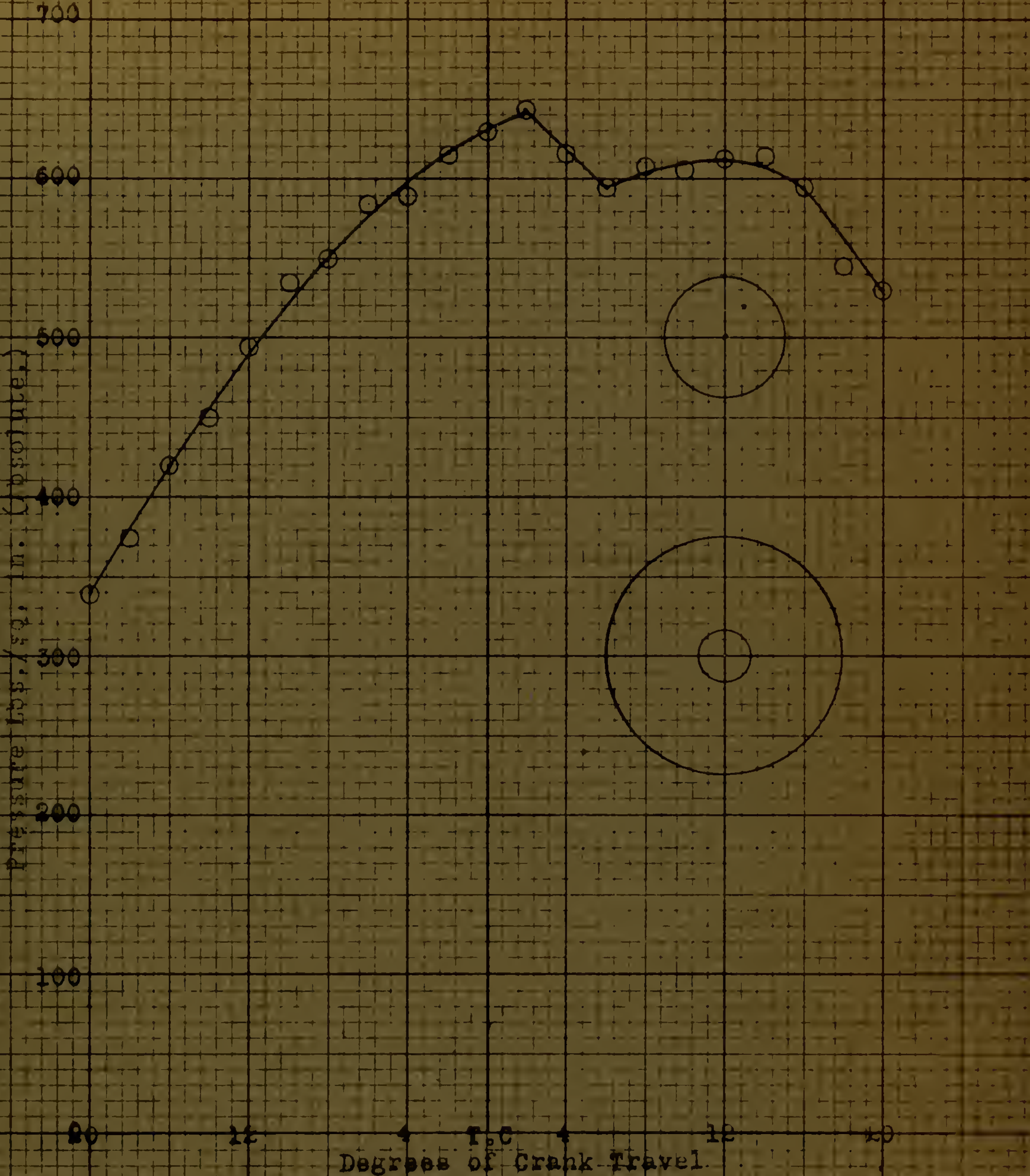
PLATE V


Curve 1.
.010" diameter Nozzle, #1 Precombustion Chamber.
Normal Advance.
Curve 2.
Normal Advance.
Percent Rated Load
.8
.6
.4
50
100

PLATE VI


Lbs. Fuel/BHP/hr.
Advanced Injection.
Advanced Injection.
25
50
75
100
Percent Rated Load
PLATE VII

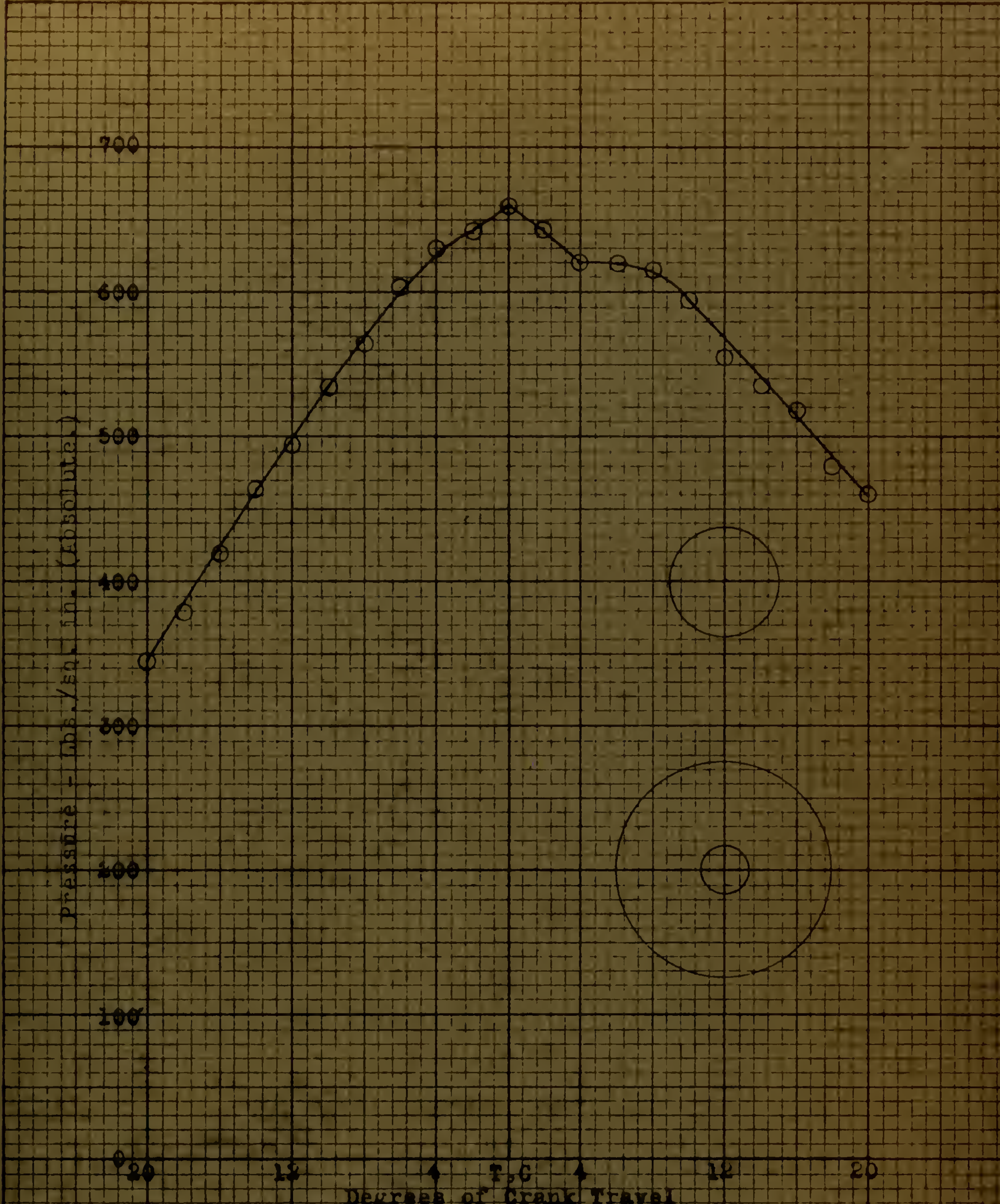

.8
.6
.4
Curve 1.
Precombustion Chamber, .020" diameter Nozzle.
Precombustion Chamber .020" diameter Nozzle.
Precombustion Chamber .020" Diameter Nozzle.
25
50
75
100
Percent Rated Load.
PLATE VIII



<u>Card #1-3.</u> Nozzle Diameter .020"
Precombustion Chamber #1.

800 R.P.M. 32 B.H.P. B.M.E.P. - 57.6 lbs/sq.in.
Economy: .505 lbs. fuel/BHP/hr.

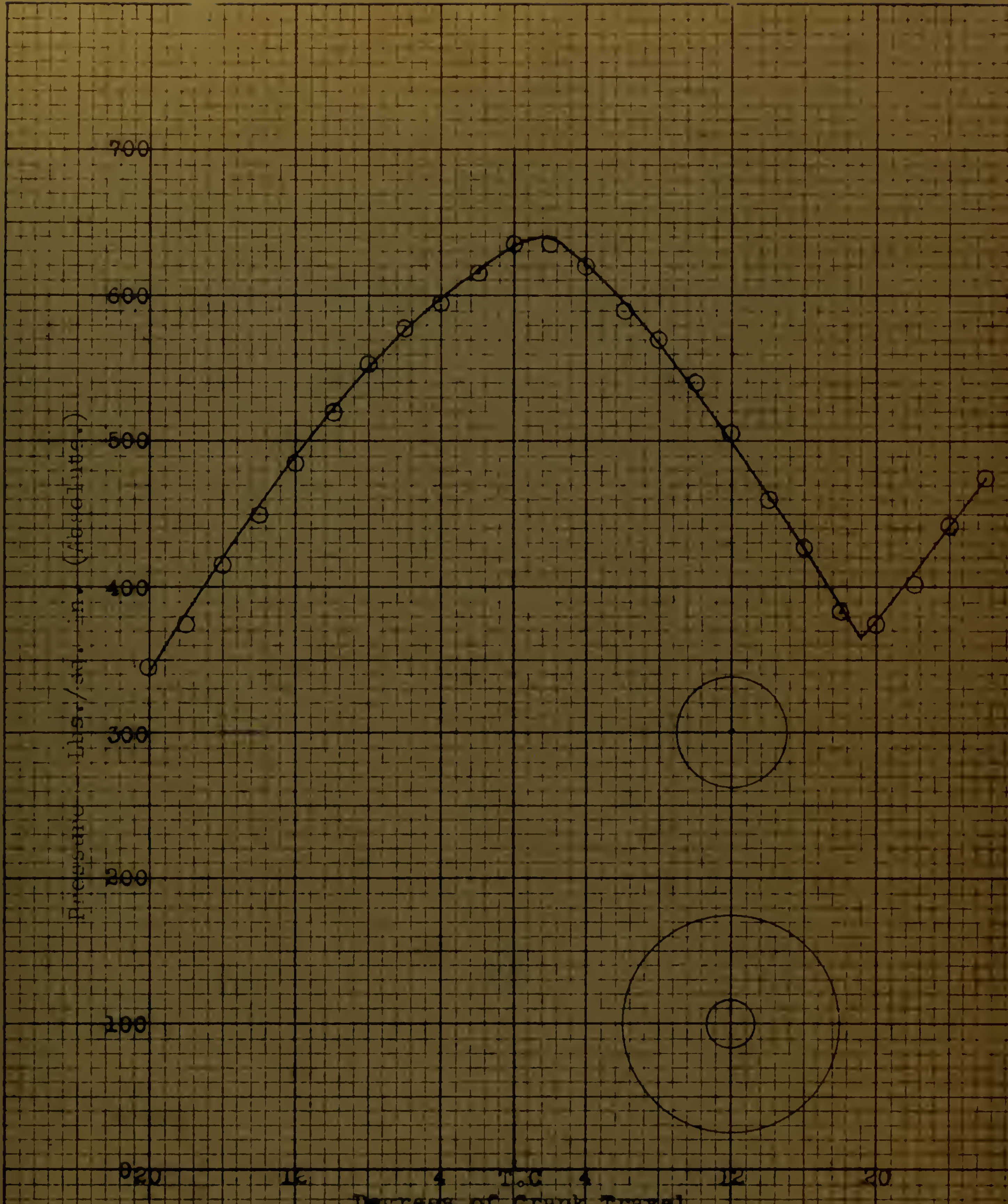
PLATE IX



Card #2-4. Nozzle Diameter .010"
Precombustion Chamber #1.

600 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs./sq.in.
Economy: .800 lbs. fuel/BHP/hr.

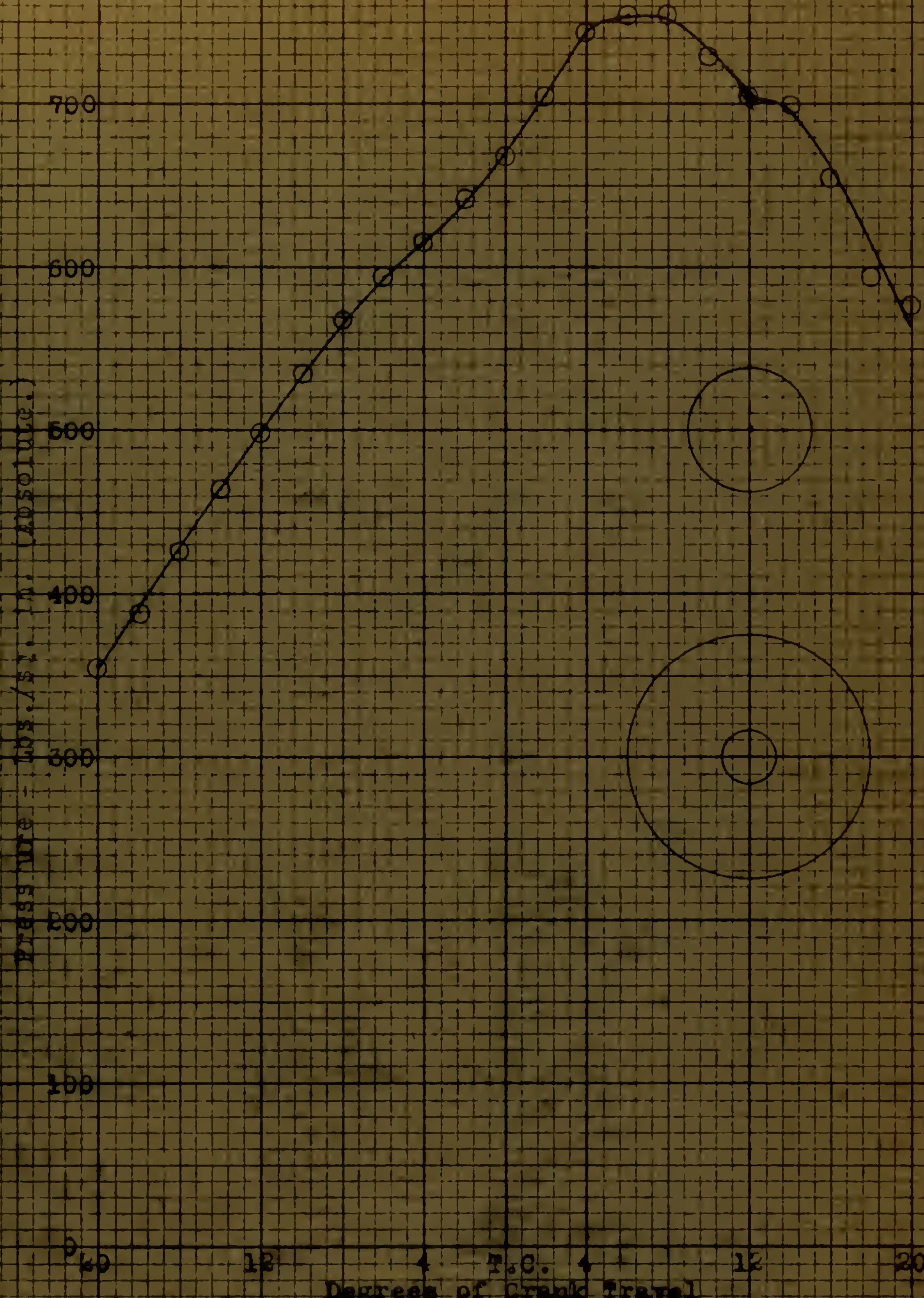
PLATE X



Card #3-2. Nozzle Diameter .030"
Precombustion Chamber #1.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs./sq.in.
Economy: .572 lbs. fuel/BHP/hr.
Normal Injection Advance.

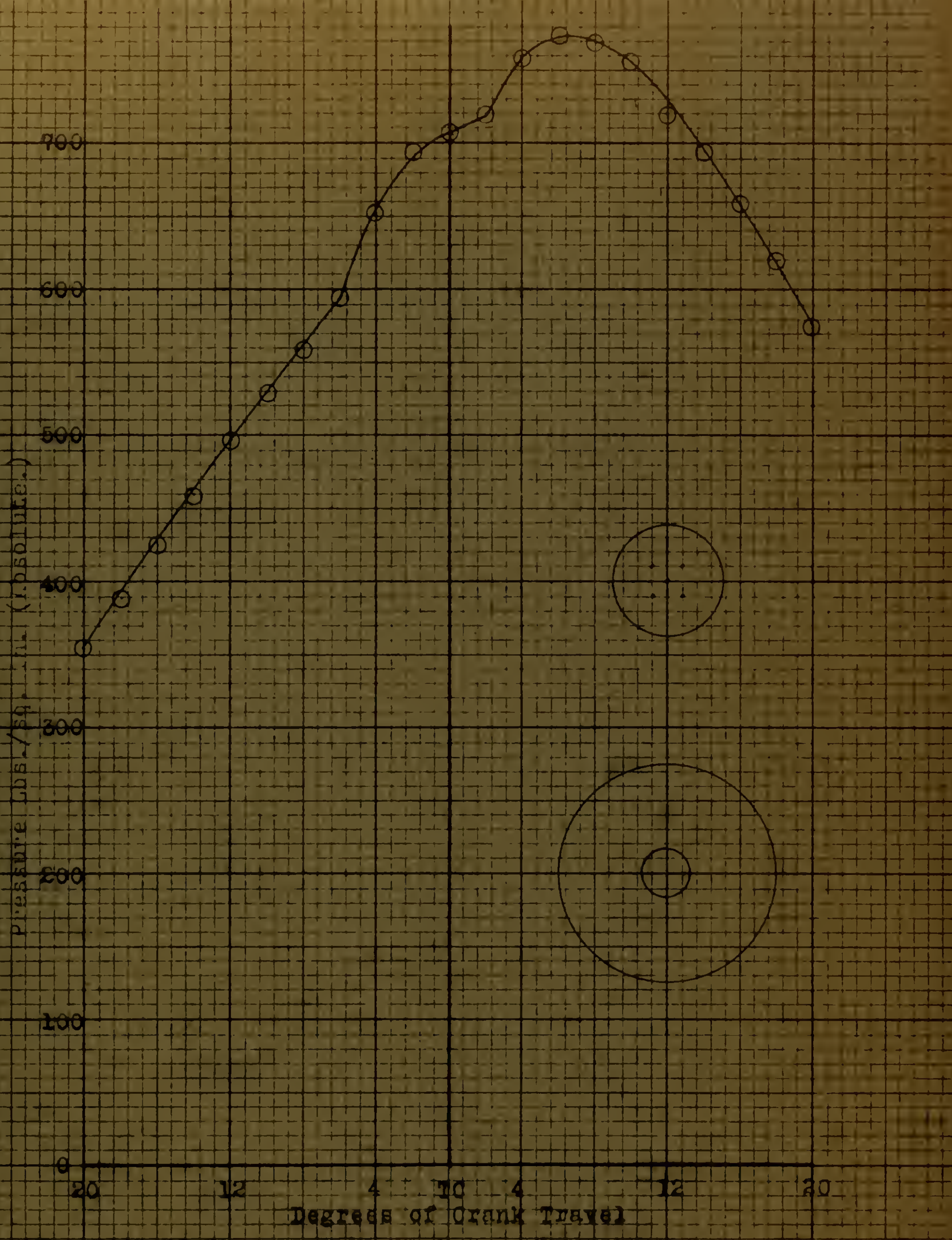
PLATE XI



Card #4-3. Nozzle Diameter .030"
Precombustion Chamber #1.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs./sq.in.
Economy: .542 lbs. fuel/BHP/hr.
Advanced Injection.

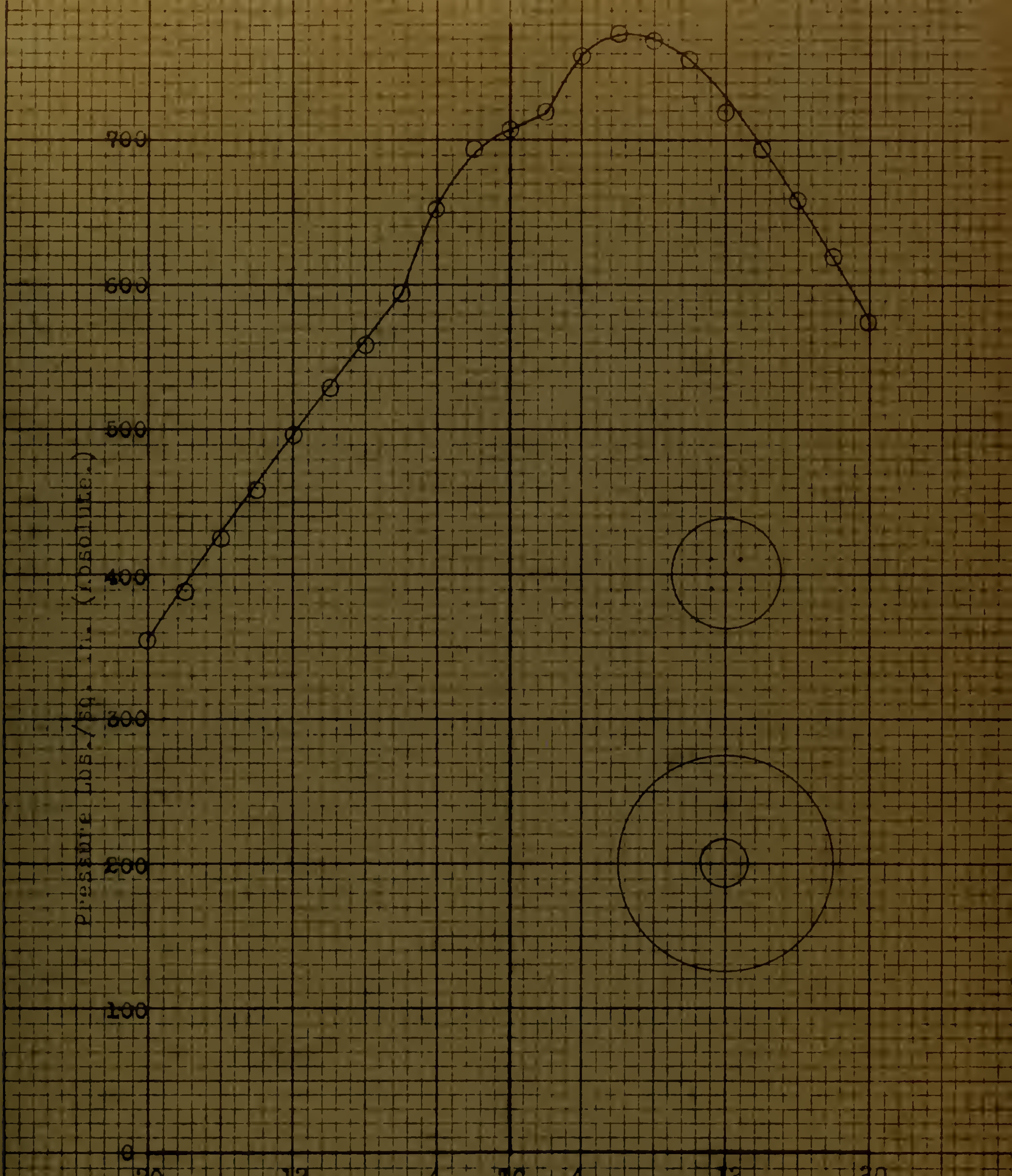
PLATE XII



Card #5-5. Nozzle Diameter .010" - 5 Orifices.
Precombustion Chamber #1.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs/sq.in.
Economy: .544 lbs. fuel/BHP/hr.
Advanced Injection.

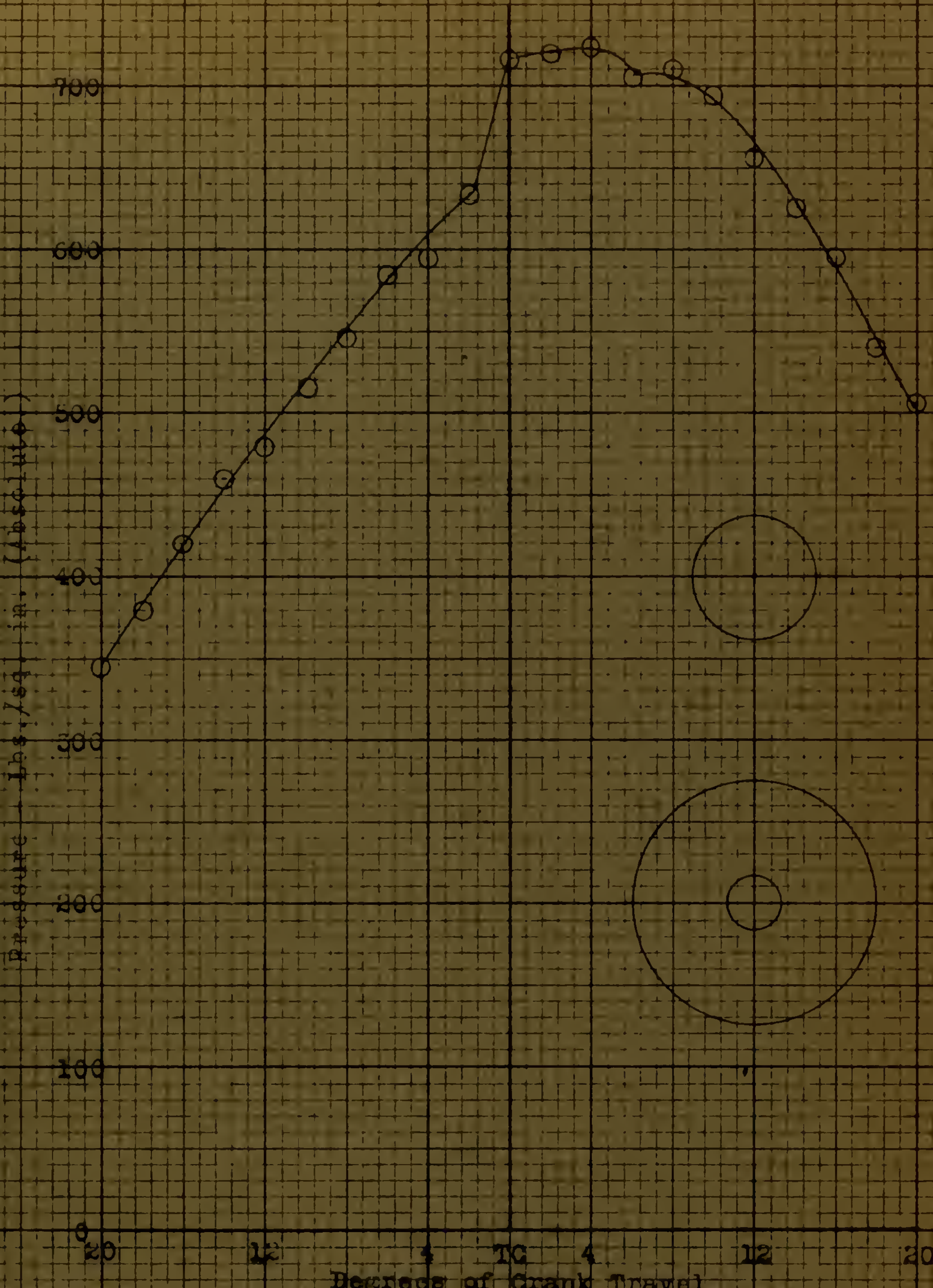
PLATE XIII



Card #5-5. Nozzle Diameter .010" - 5 Orifices.
Precombustion Chamber #1.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs/sq.in.
Economy: .544 lbs. fuel/BHP/hr.
Advanced Injection.

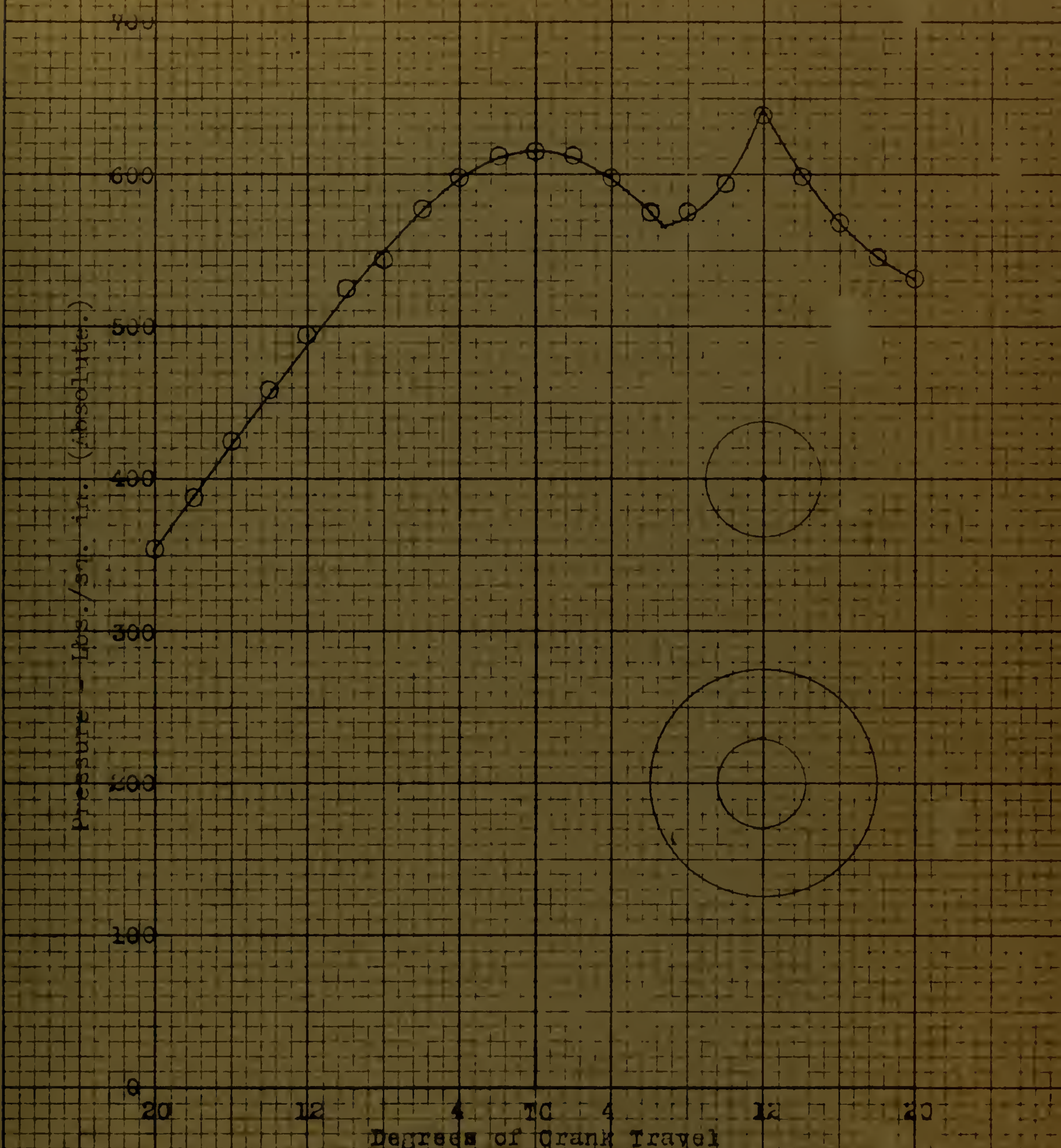
PLATE XIII



Card #5-4/ Nozzle Diameter .010"
Precombustion Chamber #1.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 56.9 lbs/sq.in.
Economy: .805 lbs fuel/BHP/hr.
Advanced Injection.

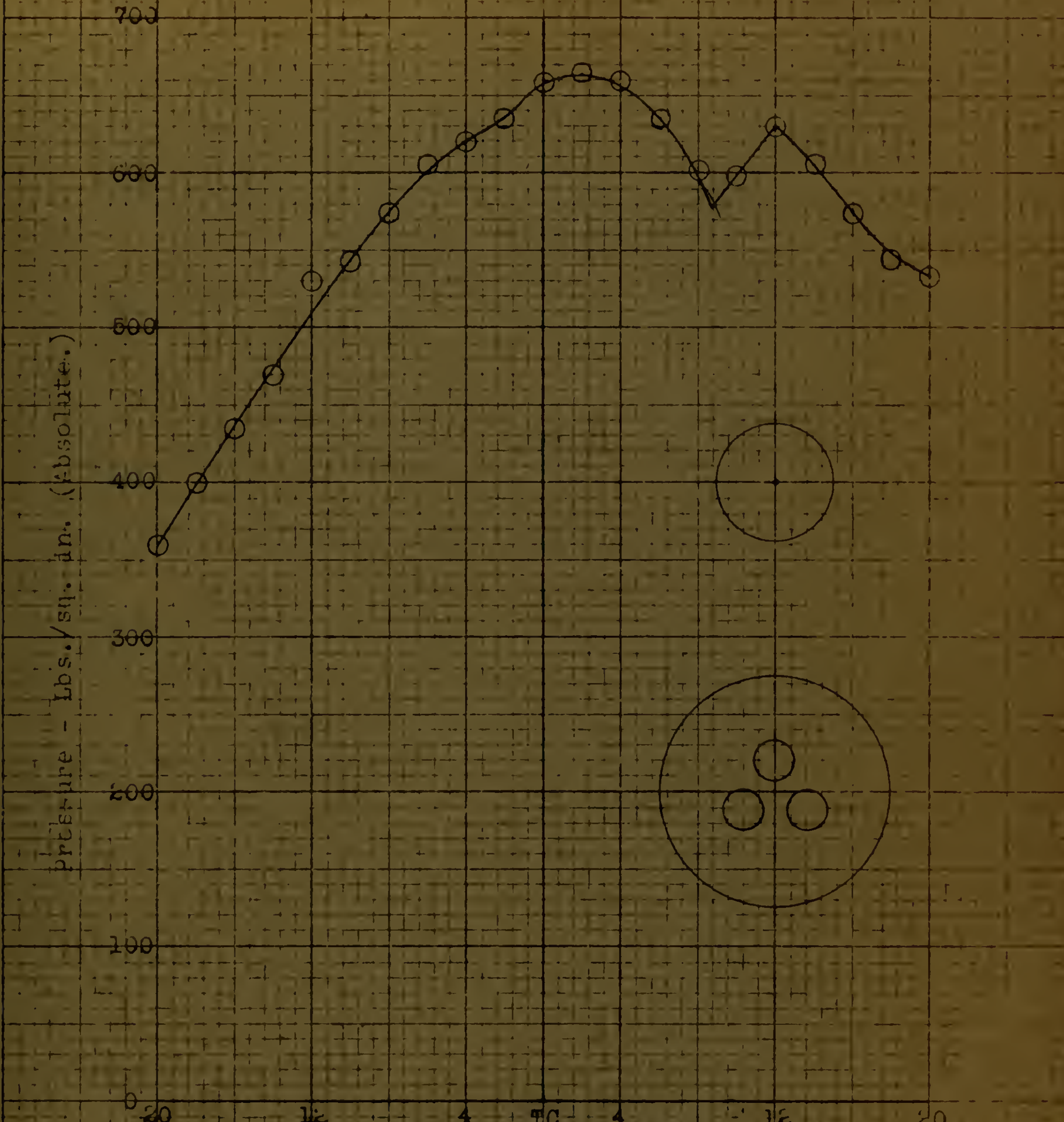
PLATE XIV



Card #7-3. Nozzle Diameter .020"
Precombustion Chamber #3.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 66.9 lbs./sq.in.
Economy: .527 lbs. fuel/BHP/hr.
Normal Injection Advance.

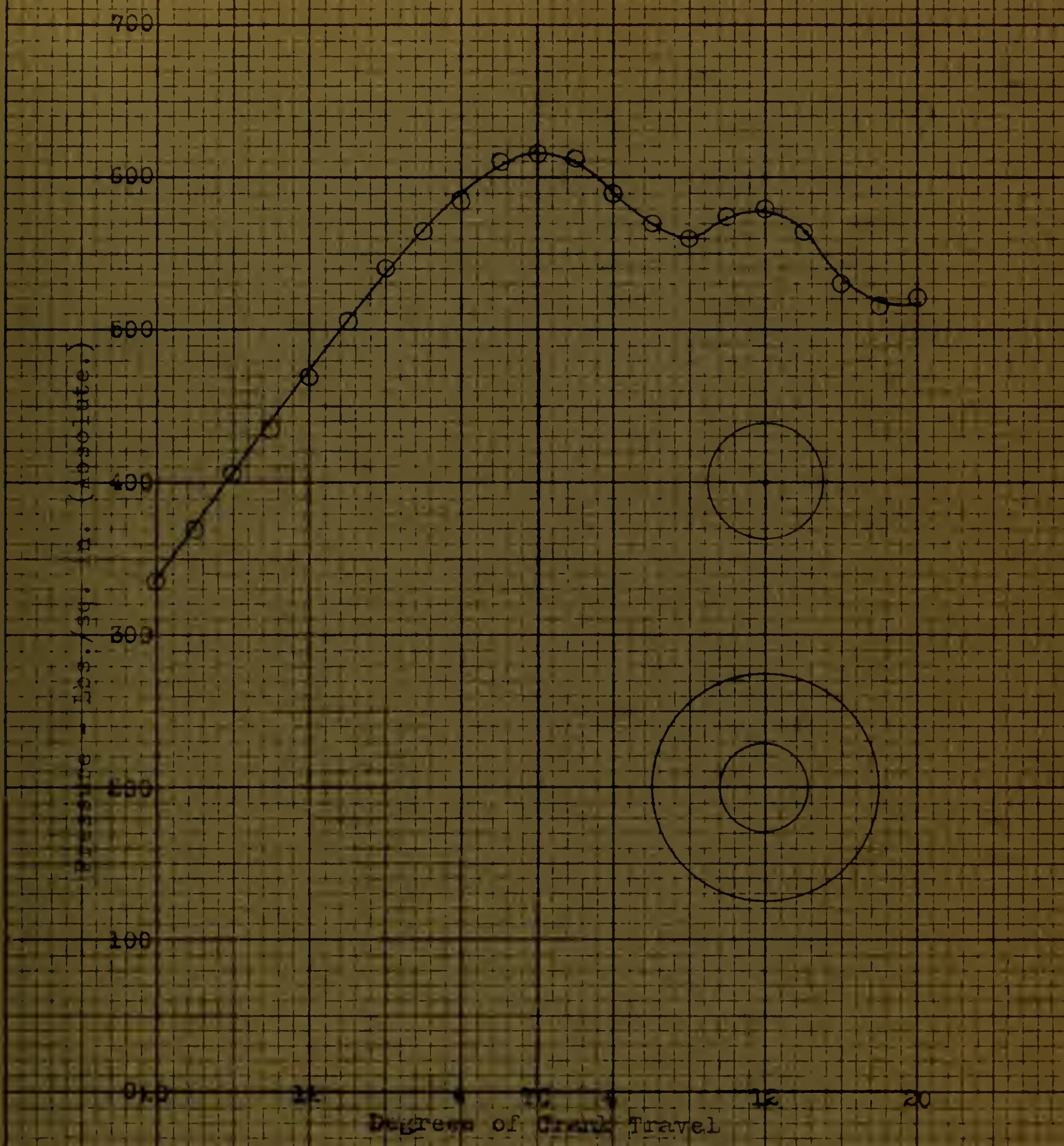
PL. XV



Degrees of Crank Travel.
Card #8-4. Nozzle Diameter .020"
Precombustion Chamber #2.

800 R.P.M. 31.6 B.H.P. B.M.E.P. - 58.9 lbs./sq.in.
Economy: .525 lbs. fuel/BHP/hr.

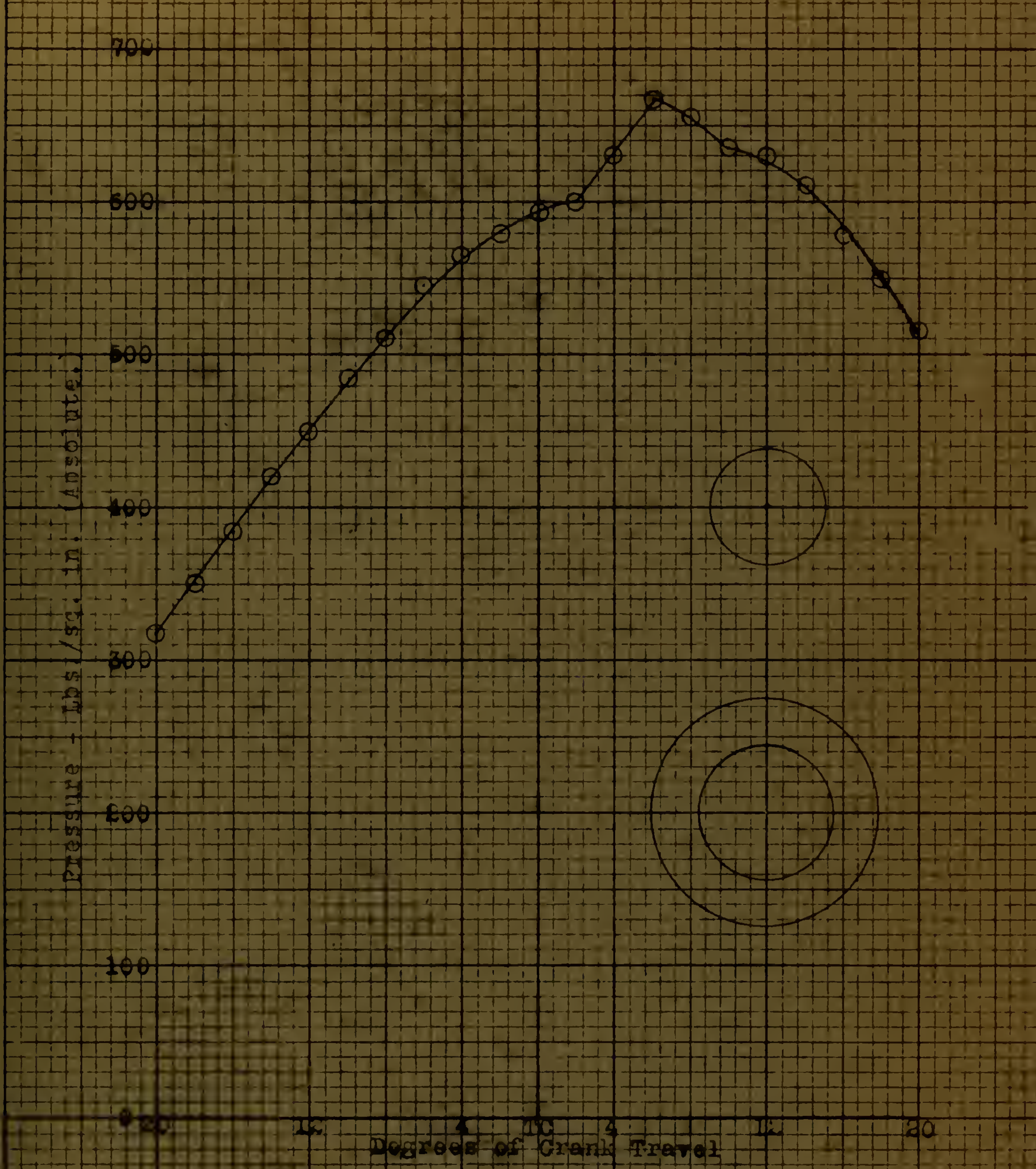
Normal Injection Advance.



Card #[illegible]. Nozzle Diameter .020"
Precombustion Chamber #3a.

800 R.P.M. [illegible] B.H.P. B.M.E.P. - 50.9 lbs/sq.in.
Economy: .228 lbs. Fuel/BHP/Hr.
Normal Injection Advance.

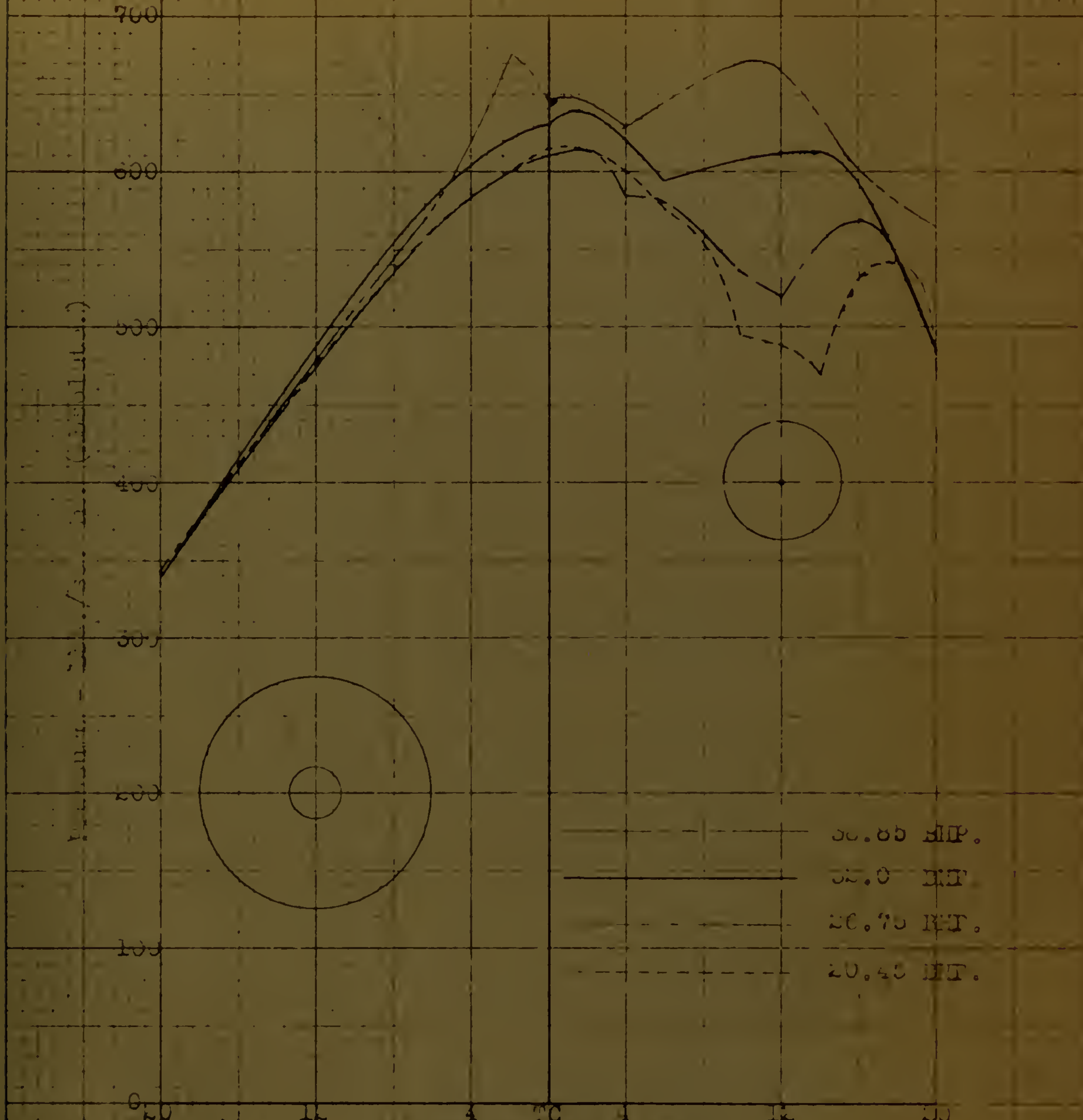
PLATE XVII



Card #10-5. Nozzle Diameter .020"
Precombustion Chamber #4.

900 R.P.M. 21.5 B.H.P. B.M.E.P. - 56.9 lbs./sq.in.
Economy: .558 lbs. fuel/BHP/hr.
Advanced Injection.

PLATE XVIII



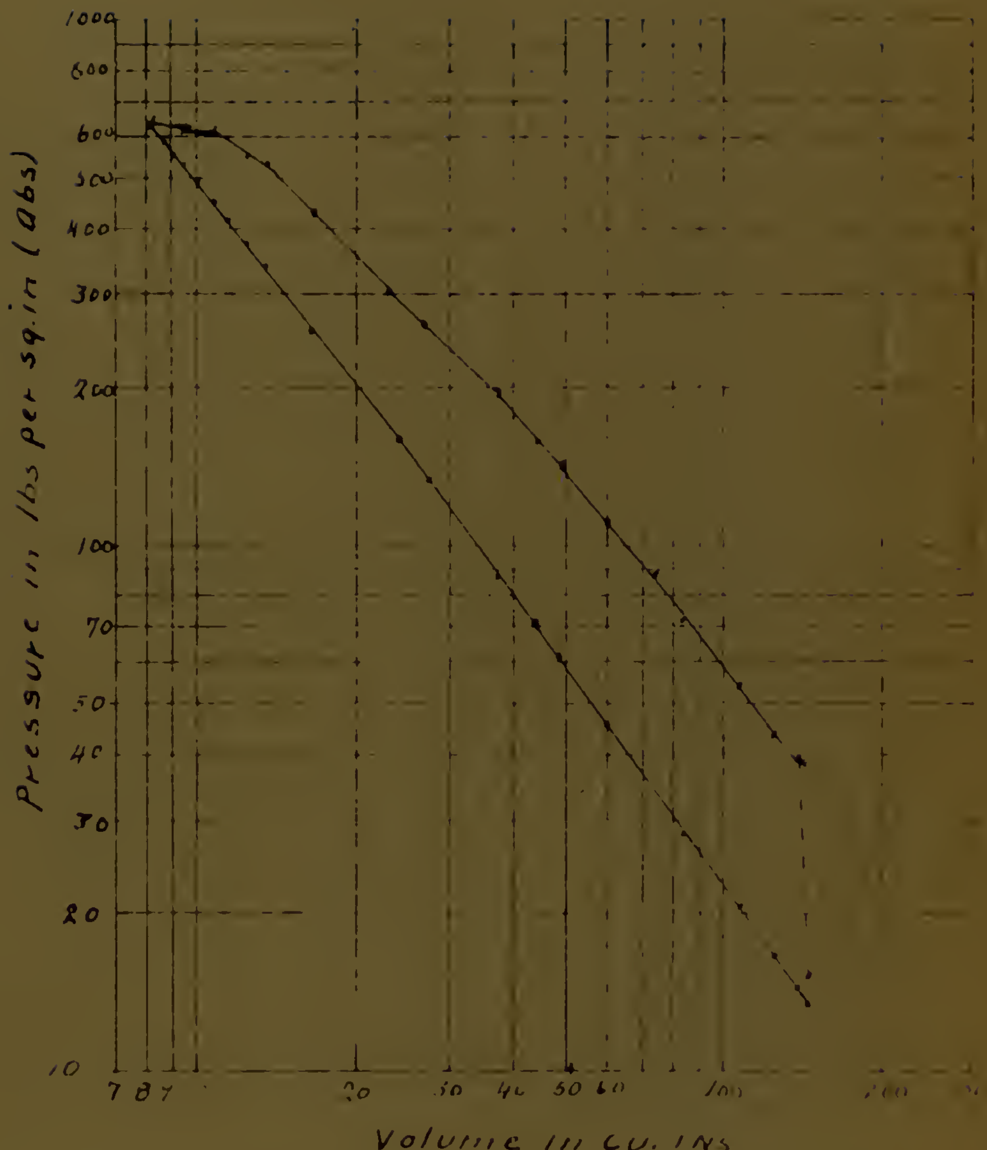
Degrees of Crank Travel.

Nozzle Diameter .020"

Precombustion Chamber #1.
800 RPM.

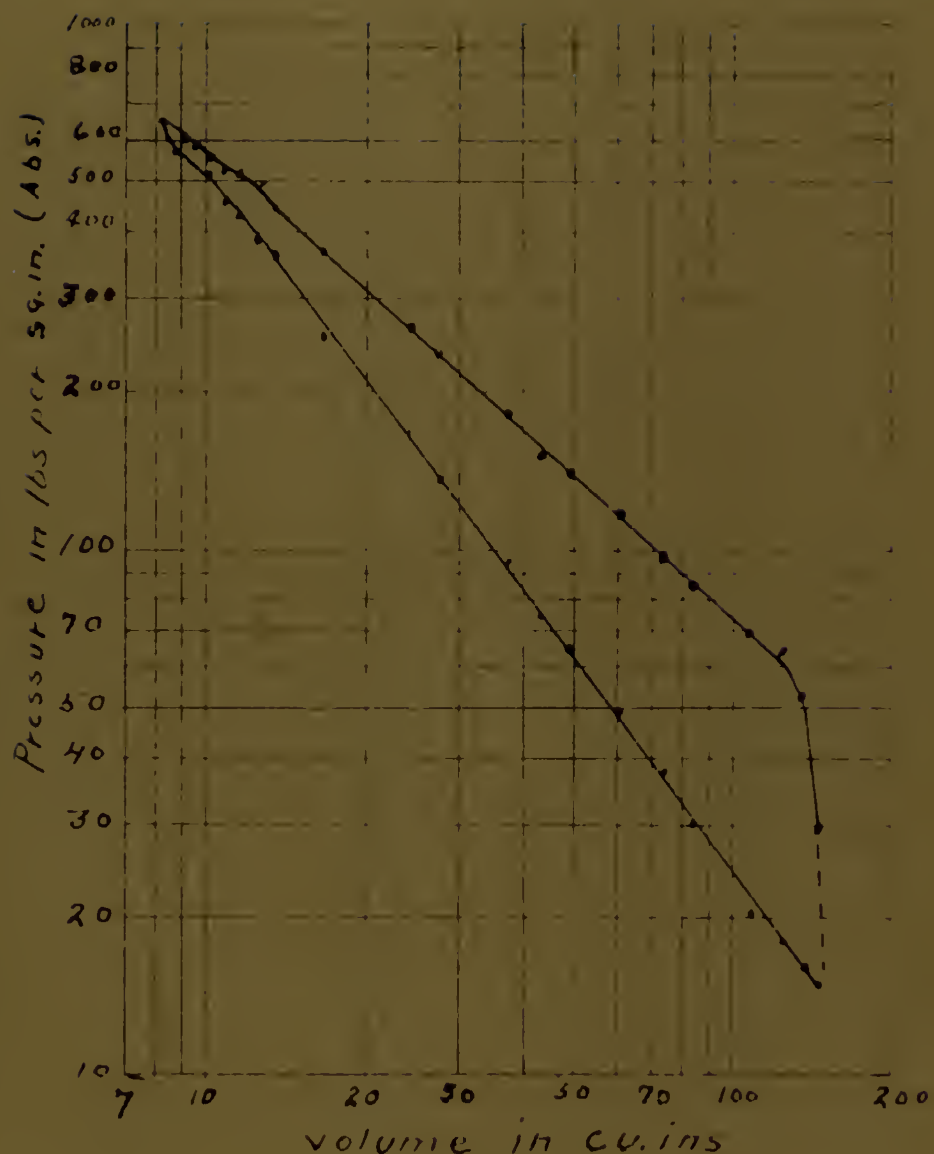
Normal Injection Advance.

PLATE XIX



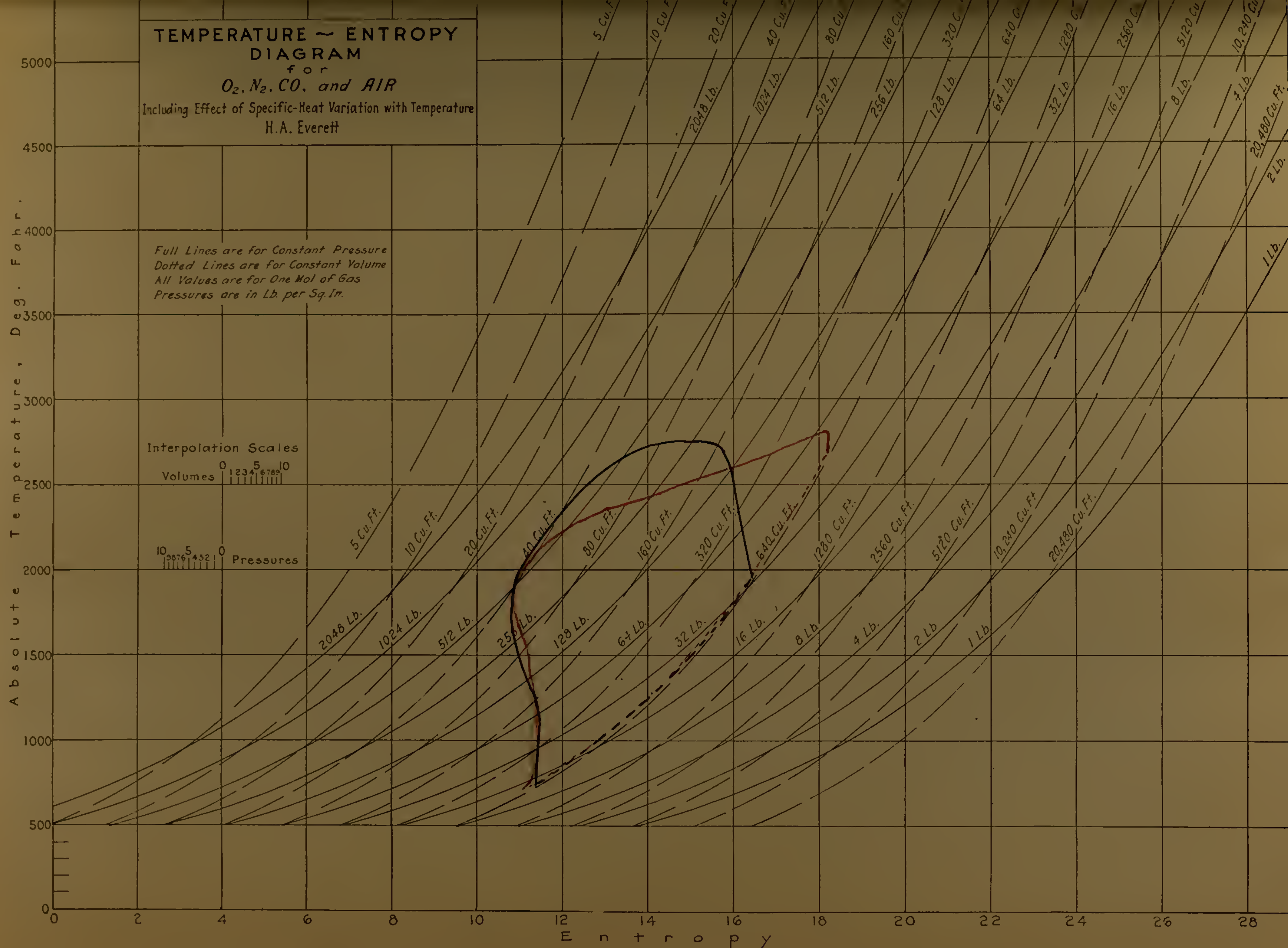
Spray Nozzle diam = .020 ins

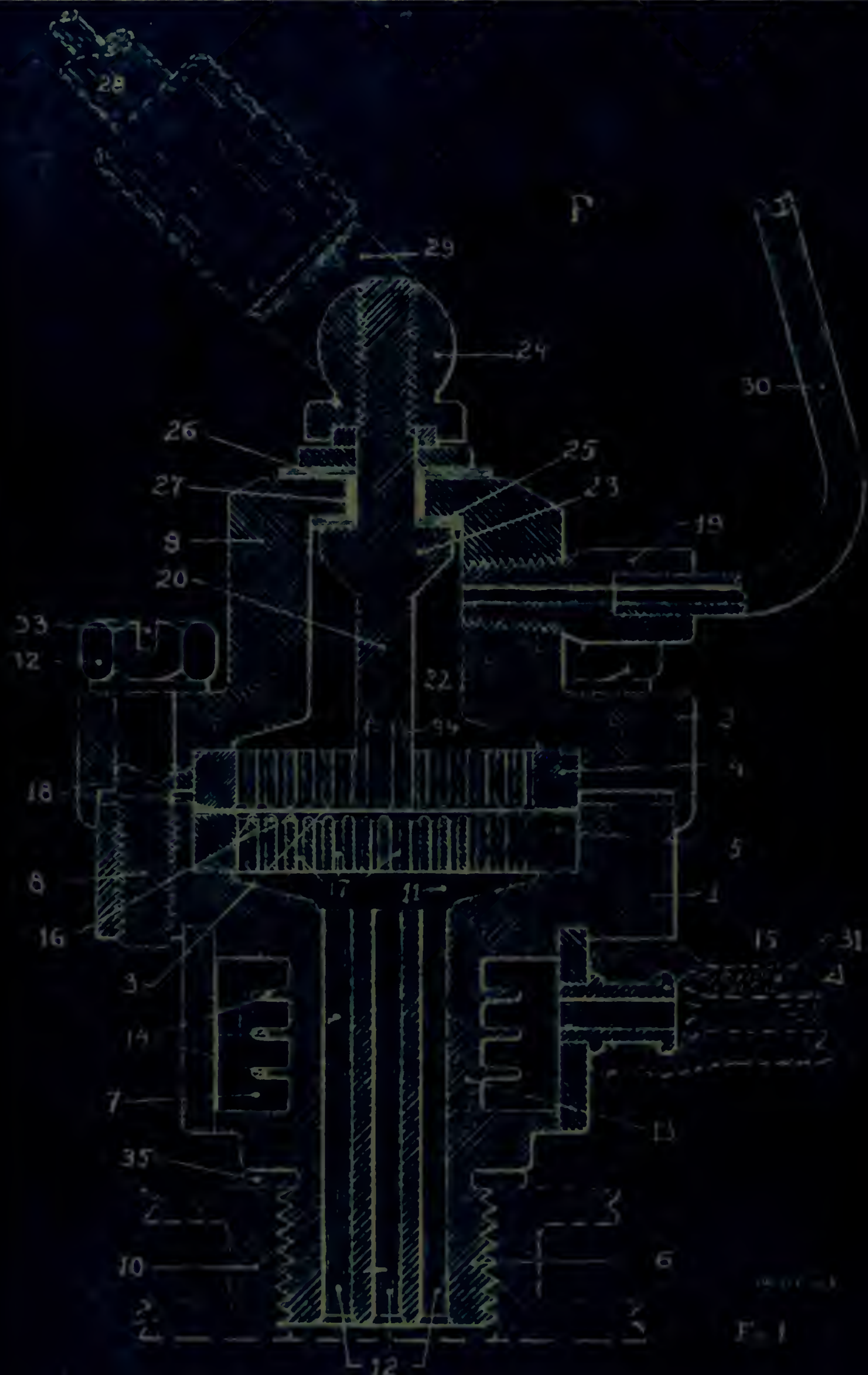
Plate XX



Spray Nozzle diam .010 in

Plate XXI


TEMPERATURE ~ ENTROPY
DIAGRAM
for
O2, N2, CO, and AIR
Including Effect of Specific-Heat Variation with Temperature
H.A. Everett
Full Lines are for Constant Pressure
Dotted Lines are for Constant Volume
All Values are for One Mol of Gas
Pressures are in Lb. per Sq. In.
Interpolation Scales
Volumes
Pressures
5 Cu. Ft.
10 Cu. Ft.
20 Cu. Ft.
40 Cu. Ft.
80 Cu. Ft.
160 Cu. Ft.
320 Cu. Ft.
640 Cu. Ft.
1280 Cu. Ft.
2560 Cu. Ft.
5120 Cu. Ft.
10,240 Cu. Ft.
20,480 Cu. Ft.
2048 Lb.
1024 Lb.
512 Lb.
256 Lb.
128 Lb.
64 Lb.
32 Lb.
16 Lb.
8 Lb.
4 Lb.
2 Lb.
1 Lb.
Absolute Temperature, Deg. Fahr.
Entropy
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28



PRESSURE INDICATOR

FRONT VIEW

TIMER FOR INDICATOR

TIMER FOR INDICATOR

Fig. 7

TABULAR DATA SHEETS

Offset Diagram Readings										
Crank Degrees	Card 1-3	Card 2-4	Card 3-2	Card 4-3	Card 5-5	Card 6-4	Card 7-3	Card 8-4	Card 9-4	Card 10-5
20	340	345	345	354	354	345	354	345	355	317
18	375	380	375	369	389	38[illegible]	389	385	370	350
16	420	4[illegible]	415	425	424	420	424	4[illegible]	405	385
14	450	465	450	464	459	45[illegible]	459	455	4[illegible]	4[illegible]
12	495	495	485	499	497	450	494	515	470	450
10	535	535	520	5[illegible]	529	515	52[illegible]	527	[illegible]5	465
8	550	565	593	53[illegible]	559	547	544	5[illegible]	54[illegible]	510
6	565	6[illegible]	578	594	594	565	579	590	5[illegible]5	545
4	590	630	595	614	652	[illegible]5	599	6[illegible]	565	565
2	625	642	615	642	694	6[illegible]3	611	62[illegible]	61[illegible]	580
0	650	660	635	669	7[illegible]	715	614	645	615	593
2	642	645	635	704	719	720	611	650	612	6[illegible]
4	615	620	620	744	759	725	599	645	590	6[illegible]
6	595	620	590	754	774	7[illegible]	574	620	570	607
8	6[illegible]7	615	570	754	679	710	576	580	56[illegible]	655
10	6[illegible]5	595	540	729	757	685	594	583	575	635
12	611	555	505	704	719	655	639	615	580	630
14	612	535	460	699	694	625	599	590	565	610
16	595	516	4[illegible]7	654	659	595	569	5[illegible]	550	578
18	545	480	383	594	619	540	544	550	515	550
20	530	450	375	577	574	505	55[illegible]	517	52[illegible]	515
26			475							

All pressures corrected for barometer, and zero indicator reading

DATA SHEET 1

Date	Run No.	[illegible]	[illegible]	[illegible]
	1	[illegible]7.	.7[illegible]5	
	2	22.[illegible]	.[illegible]17	
March 7	3	32.	.[illegible]	
	4	39.1	.[illegible]	[illegible]
	1a	[illegible].8	.[illegible]	
March 18	2a	[illegible].9	.[illegible]	
	3a	41.55	.[illegible]	
	1	2[illegible].8	.[illegible]	
	2	[illegible].1	.[illegible]	
April 5	3	27.4	.[illegible]	[illegible]
	4	[illegible]	.[illegible]00	
	5	[illegible].7	.[illegible]	
	1	3[illegible].9	.[illegible]77	
	2	31.[illegible]	.[illegible]7[illegible]	
April 12	3	27.4	.[illegible]	[illegible]
	4	[illegible].1	.[illegible]	
	5	2[illegible].3	.[illegible]7[illegible]	
	1	[illegible].1	.[illegible]	
	2	[illegible]	.[illegible]	
April 15	3	[illegible]	.5[illegible]	[illegible]
	4	2[illegible]	.[illegible]	
	5	[illegible].9	.[illegible]	
	1	1[illegible]	.7[illegible]	
	2	[illegible]	.[illegible]	
	3	[illegible].1	.[illegible]	
April 19	4	[illegible]7.[illegible]	.[illegible]	[illegible]
	5	[illegible]	.[illegible]	
	6	[illegible].9	.[illegible]	

DATA SHEET 2

Date	Run No.	[illegible]	[illegible]	[illegible]
April 8	1	[illegible].1	[illegible]	[illegible]
	2	[illegible]	.7[illegible]7	
	3	27.[illegible]	.742	
	4	[illegible]	[illegible]	
April 20	1	[illegible].9[illegible]	[illegible]	7
	2	[illegible].9[illegible]	[illegible]	
	3	[illegible]	.527	
	4	[illegible].1	[illegible]	
	5	[illegible].9	.517	
	6	27.4	[illegible]	
	7	22.4	.590	
April 22	1	[illegible].8	[illegible]	[illegible]
	2	[illegible].1	[illegible]	
	3	[illegible].4	[illegible]	
	4	[illegible]	[illegible]	
April [illegible]	1	[illegible].8	[illegible]	[illegible]
	2	[illegible].1	[illegible]	
	3	27.4	[illegible]1	
	4	[illegible]	[illegible]8	
	5	34.9	[illegible]7	
April 26	1	[illegible].1	[illegible]	[illegible]
	2	[illegible].1	[illegible]	
	3	[illegible].8	.598	
	4	[illegible]7.4	[illegible]	
	5	[illegible]	[illegible]58	
	6	[illegible].9	[illegible]5	

DATA SHEET 3

PRESSURE, VOLUME AND TEMPERATURE RELATIONSHIPS

FOR .010 INCH NOZZLE

Degrees from Top D. C.	Fraction of Stroke	Displacement Volume + Clearance	$\frac{Vol}{Mass} = V$	[illegible] V	Pressure Compression Stroke	Temperature Compression Stroke	Pressure Expansion Stroke	Temperature Expansion Stroke
0	0	8.25	1.155	33.5	645	2020	6[illegible]5	[illegible]
2	.0004	[illegible]	1.15	33.51	620	1950	635	1900
4	.0015	8.450	1.18	34.2	612	1950	620	1900
6	.0034	8.718	1.22	35.4	585	1930	585	1930
8	.0060	9.075	1.27	36.8	560	1920	565	[illegible]
10	.0092	9.52	1.33	38.0	550	1900	575	2[illegible]40
12	.0135	10.11	1.415	41.0	502	1920	555	[illegible]
14	.0182	10.75	1.505	45.6	400	1870	525	2100
16	.0238	11.52	1.61	46.6	425	1850	510	22[illegible]0
18	.0300	12.38	1.73	50.2	385	1800	495	2310
20	.0367	13.30	1.86	54.0	360	16[illegible]0	440	2210
26	.0595	16.33	2.28	66.1	250	1570	365	2250
30	.114	23.95	3.35	97.1	165	1545	265	2400
40	.140	27.50	3.85	111.5	137	1400	235	2540
50	.2115	37.35	5.22	151.5	94	1220	180	2530
56	.2600	44.05	6.17	179.	74	1175	150	25[illegible]0
60	.2920	48.35	6.70	193.0	65	1120	140	25[illegible]0
70	.3746	60.35	8.45	244.1	48	1065	117	[illegible]580
80	.4077	72.05	10.2	295.	38	1035	97	2670
90	.5563	84.75	11.8	342.	30	910	85	2700
110	.7200	107.45	15.01	436.	20	832	59	2820
130	.8542	124.25	17.4	505.0	18	775	50	2820
150	.9409	138.45	19.4	564.	16	760	52	2720
180	1.000	145.75	20.3	593.	15	740	29	

DATA SHEET 4

PRESSURE, VOLUME AND TEMPERATURE RELATIONSHIPS

FOR .080 INCH NOZZLE.

Degrees from Top D. C.	Fraction of Stroke	Displacement Volume + Clearance	Vol / [illegible] = V	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
0	0	8.[illegible]	1.115	[illegible]	[illegible]	13[illegible]	[illegible]	[illegible]
2	.0004	8.[illegible]	1.1[illegible]	[illegible]	[illegible]	1[illegible]	[illegible]	[illegible]
4	.0015	8.4[illegible]	1.1[illegible]	[illegible]	[illegible]	1[illegible]	14.[illegible]	1[illegible]
6	.0034	8.71[illegible]	1.[illegible]	75.4	[illegible]	13.5	[illegible]	1[illegible]
8	.0060	9.07[illegible]	1.[illegible]7	[illegible].6	[illegible].5	1[illegible]	[illegible]	[illegible]
10	.0095	9.[illegible]	1.[illegible]	38.6	[illegible].5	1[illegible]	[illegible]	1[illegible]
12	.0135	10.11	1.41[illegible]	[illegible]1.0	[illegible]	1[illegible]	11.[illegible]	[illegible]
14	.0138	10.7[illegible]	1.[illegible]	4[illegible]	[illegible]	1[illegible]	[illegible]	[illegible]
16	.0228	11.[illegible]	1.[illegible]1	[illegible]	41[illegible].5	1[illegible]	[illegible]	[illegible]
18	.0300	1[illegible]	1.[illegible]7	[illegible]	[illegible]	17[illegible]0	[illegible]	[illegible]
20	.0367	1[illegible]	1.[illegible]	[illegible]4.0	[illegible]	170[illegible]	[illegible]	[illegible]
26	.0595	1[illegible]	[illegible]	[illegible].1	[illegible]	1[illegible]	[illegible]	[illegible]
36	.114	[illegible]	[illegible]	97.1	1[illegible]	[illegible]	[illegible]	[illegible]
40	.140	[illegible]	[illegible]	111.[illegible]	1[illegible]	1[illegible]00	[illegible]	[illegible]
50	.211[illegible]	[illegible]	[illegible]	1[illegible]	[illegible]	1[illegible]	1[illegible]	[illegible]
56	.2600	[illegible].0[illegible]	[illegible].17	17[illegible]	[illegible]0.[illegible]	1[illegible]	1[illegible]	[illegible]
60	.2920	[illegible]	[illegible].7[illegible]	1[illegible].0	[illegible]1.[illegible]	11[illegible]	1[illegible]	[illegible]
70	.3786	[illegible]	[illegible]	[illegible].1	4[illegible]	10[illegible]	111.	[illegible]
80	.4677	7[illegible].6[illegible]	10.[illegible]	[illegible]	[illegible]	10[illegible]	[illegible]	[illegible]70
90	.5567	[illegible]	11.[illegible]	[illegible]	[illegible]	910	[illegible]	[illegible]20
110	.7206	10[illegible]	1[illegible].01	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
130	.8542	1[illegible]4.[illegible]	17.[illegible]	[illegible].0	[illegible]	[illegible]	[illegible]	[illegible]
150	.9469	1[illegible]	19.4	[illegible]	1[illegible]	7[illegible]0	[illegible]	[illegible]
180	1.000	1[illegible].7[illegible]	[illegible]	[illegible]30.	1[illegible]	[illegible]	1[illegible]	[illegible]